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Technical Evaluation of Combined Gas Turbine and Steam Turbine Propulsion System for Canadian Arctic Icebreaking Duty

Summary Report

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March 1983

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TECHNICAL EVALUATION OF COMBINED GAS TURBINE AND STEAM TURBINE PROPULSION SYSTEM FOR CANADIAN ARCTIC ICEBREAKING DUTY

SUMMARY REPORT

BY

ARCTIC PILOT PROJECT, INC.
CALGARY, ALBERTA

MARCH 1983

The contents of this report reflect the views of the Arctic Pilot Project and not necessarily the official views or policies of the Transportation Development Centre, Transport Canada.

PROJECT TEAM

The project team, from the Arctic Pilot Project, consisted of:

Mr. E. W. Thompson,
Mr. J. G. Wainwright,
Mr. P. J. Wyld,
and others.

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17. Abstract

This study addresses the technical suitability, for installation aboard a Canadian Arctic icebreaking vessel, of a propulsion system consisting of aero-derived gas turbines, waste heat boilers and steam turbines. In particular, the study addresses the ability of this propulsion system to:

- withstand rigorous icebreaking duties;
- deliver maximum power when required;
- perform reliably in an icebreaking application.

This study develops a preliminary design for combined cycle system suitable for an Arctic icebreaker. The study evaluates the ability and performance of this system in conducting icebreaking manoeuvres by analyzing the results of computer-based simulations. The study assesses the reliability of combined cycle propulsion for an icebreaking vessel through evaluation of experience of land-based combined cycle systems, marine wate heat boiler systems and marine high pressure steam propulsion systems. The study concludes that a combined cycle propulsion system appears to be technically suitable for installation in a Canadian Arctic icebreaking vessel.

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16 Agent de projet — CDT
I.M. Bayly

17 Resume

Ce rapport traite de l'étude de faisabilité concernant la mise en place à bord d'un navire brise-glace canadien d'un système propulsif comprenant des turbines à gaz dérivées des turbomachines pour avions, des chaudières de récupération et des turbines à vapeur. L'étude a porté plus particulièrement sur l'aptitude d'un tel système à:

- résister aux conditions rigoureuses de service
- fournir au moment voulu la puissance maximale exigée
- fonctionner fiablement dans les opérations de déglaçage.

L'étudea dégagé le concept d'un système propulsif combinant les éléments précités et convenant aux opérations dans l'Arctique, concept dont elle évalue les caractéristiques et les performances par l'examen du résultat de simulations faites à l'aide d'un ordinateur; elle en évalue aussi la fiabilité dans le contexte d'un navire brise-glace, à la lumière de l'expérience acquise dans les applications terrestres des systèmes propulsifs de cette nature et dans l'exploitation de systèmes de propulsion navale à chaudières de récupération et à turbines à vapeur haute pression. Enfin, l'étude conclut qu'un système propulsif du type proposé serait techniquement envisageable pour un navire brise-glace de cote arctique opérant dans les eaux canadiennes.

18. Mots-clés
Navire brise-glace, navire de cote arctique, cycle propulsif, turbines à gaz et à vapeur, chaudières de récupération

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Les organismes parraines dispose d'un nombre limité d'exemplaires

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EXECUTIVE SUMMARY

In early 1974, the influence of the Organization of Petroleum Exporting Countries (OPEC) cartel was first seen as the price of the benchmark crude oil, Saudi Arabian light, climbed almost 300% to $9.60 per barrel. Since this auspicious beginning, OPEC has continued to exert its strength. In the two years between January 1979 and January 1981, the contract price of Saudi Arabian light crude oil was increased eight times as the per barrel value climbed 240% from US $13.34 to US 32.00. In October 1981, the contract price of Saudi Arabian light crude oil was increased to US $34.00 per barrel where it remains in late 1982.

As fuel prices climb, it becomes economical to install equipment which captures energy that otherwise might be lost. For instance, with equipment that rejects a large quantity of heat in the exhaust gases, such as diesel engines and gas turbines, rising fuel prices may make the installation of waste heat boilers economical.

In 1978 and 1979, Pratt and Whitney Canada Ltd. conducted a study for the Department of Transport, which investigated the feasibility of installing in a marine application a propulsion system consisting of aero-derived gas turbines equipped with waste heat boilers, to produce steam for additional propulsive power. This study concluded that it is feasible to install a combined gas turbine and steam turbine propulsion system in a marine application.

This report supplements the initial studies conducted by Pratt and Whitney by assessing the technical feasibility of installing a combined gas turbine and steam turbine propulsion system in a Canadian Arctic icebreaker specifically the liquid natural gas carriers, proposed by the Arctic Pilot Project and being designed to Class 7 of the Arctic Shipping Pollution Prevention Regulations. This study addresses, in particular, the ability of the combined cycle system:

- to withstand rigorous icebreaking duties;
- to deliver maximum power when required;
- to perform reliably in an icebreaking application.

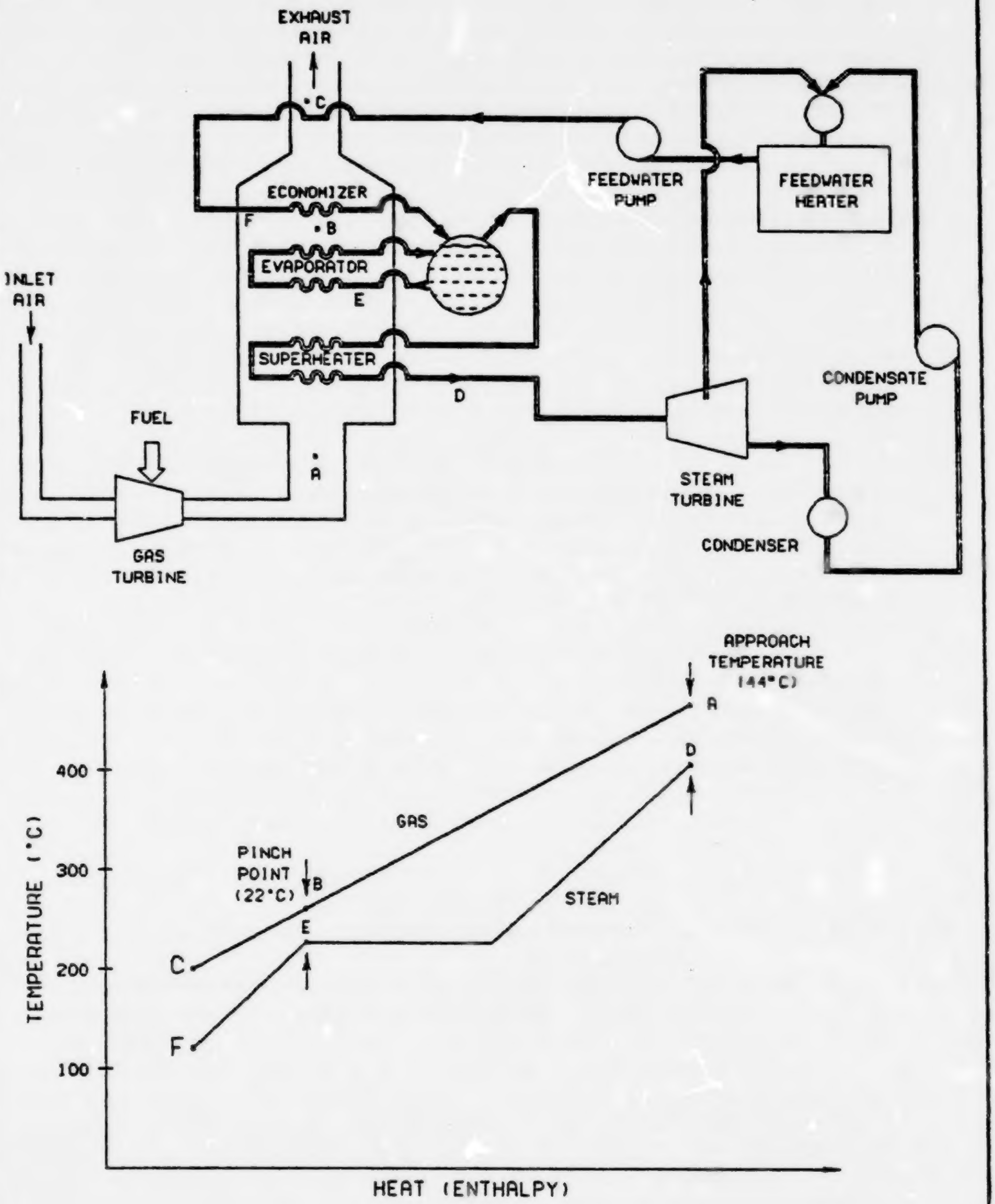
A combined cycle system suitable for a Canadian Arctic icebreaker is depicted in Figure 1. The system might be comprised of Rolls Royce RB-211 or General Electric LM 2500 aero-derived gas turbines which are "second generation" gas turbines having high firing temperature and low specific fuel consumption relative to alternative gas turbines. Additionally, these are light-weight machines being able to accept the rapid load changes that occur with icebreaking.

A waste heat boiler suitable for Canadian Arctic icebreaking application might have the following characteristics:

- an approach temperature of 44°C (80°F) which represents a reasonable compromise between steam cycle efficiency and boiler capital cost
- a pinch point of 22°C (40°F) which also represents a reasonable compromise between efficiency and capital cost
- a minimum exhaust gas stack temperature of 126.7°C (260°F) to avoid the sulphur dew point and potential corrosion of boiler tubing
- superheated steam conditions at boiler outlet of approximately 400°C (752°F) temperature and 2.53 MN/m^2 (25 atmosphere) pressure
- single pressure boiler to ease control complexity
- no supplementary firing to avoid additional capital cost and simplify boiler operation
- capability of dry firing to allow gas turbine to operate when boiler is "off-line"
- one boiler for each gas turbine exhaust

The steam turbine suitable for Canadian Arctic icebreaking application will be designed with the objectives of simplicity and reliability with particular attention towards accepting "wet" steam which may be produced during operation in low ambient temperatures.

The steam system of the COGAS plant might be controlled through constant pressure at the superheater outlet. This system of control might be selected since the steam generated by the waste heat boilers will not be dedicated solely to the main propulsion system. This steam will also be used for auxiliary purposes such as ship service power using a turbo-generating set. In this situation, the ship service turbo-generators will be the preferred steam customer with the balance of the steam passed to the main propulsion turbines. The ship service turbine requires a reasonably constant steam pressure to maintain turbine speed and load.


EXHAUST
AIR
C
FEEDWATER
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HEATER
ECONOMIZER
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STEAM
TURBINE
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APPROACH
TEMPERATURE
(44°C)
A
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PINCH
POINT
(22°C)
B
300
STEAM
E
C
200
F
100
TEMPERATURE (°C)
HEAT (ENTHALPY)

FIGURE 1
TEMPERATURE-ENTHALPY DIAGRAM
OF COMBINED GAS TURBINE
AND STEAM TURBINE SYSTEM

The technical suitability of a combined cycle propulsion system for a Canadian Arctic icebreaker might be gauged by computer simulation of the equipment. Hybrid digital and analogue computers might be used to simulate the combined cycle system undertaking icebreaking manoeuvres. By monitoring the response of various aspects of the system, certain insight as to the ability of the system to undertake the anticipated duty is provided. Unfortunately, detailed dynamic simulation of the COGAS system is outside the financial bounds of this study. However, Solar Turbines International of San Diego, California have developed a combined cycle model capable of simulating the General Electric LM 2500 gas turbine and a waste heat boiler of their own design. Solar agreed to simulate, with this model and for this study, a combined cycle propulsion system conducting an ice ramming cycle perhaps the most arduous of icebreaking operations.

The Solar waste heat boiler is a two circuit, "once through", dual pressure, finned tubed boiler. The steam system is designed to operate as a "slave" to the gas turbine with boiler pressure allowed to vary with gas turbine power. The water/steam cycle is essentially closed loop with minimal make-up feedwater required.

The ramming cycle is considered onerous for an icebreaking vessel since the prime movers, in a limited period of time, complete four excursions from a low power to full power situation (refer Figure 2). For a COGAS propulsion system, this raises concerns such as:

- thermal cycle fatigue in gas turbine, waste heat boiler or steam turbine
- ability of boiler to generate steam rapidly as gas turbine power increases
- internal operation of the boiler during gas turbine power fluctuation
- ability of the propulsion system to complete the ramming cycle in the time specified due to lagging steam power.

Unfortunately, the Solar model was not sufficiently sophisticated to accept a ramp function input representing the gas turbine power. The input was therefore a step function, as shown in Figure 2, which creates a more arduous duty for the waste heat boiler. The simulation outputs monitors were solely waste heat boiler characteristics and consisted of:

- feedwater flow
- location of dryout zone (position where steam first becomes saturated)
- high pressure steam temperature
- high pressure steam flow
- low pressure steam temperature
- low pressure steam flow

Figure 2 describes the feedwater flow to the boiler and the steam flow at the outlet of the superheater. The feedwater flow in the Solar "once-through" boiler is predicated upon the power level and predicted exhaust gas characteristics of the gas turbine. With an immediate drop in gas turbine power, there is a corresponding decline in feedwater flow. The flow is maintained at this level until gas turbine power is again increased.

As feedwater flow is reduced with the initial decrease in gas turbine power, both high pressure and low pressure steam flow from the boiler increase as the pressure relief valve at the outlet strives to maintain pressure drop across the boiler. The steam flow eventually declines below the full power production. With increase in gas turbine power to full load, there is a slight lag in steam flow before increasing and then reaching equilibrium approximately one minute later.

Figures 3 and 4 describe the variation in low pressure and high pressure steam flow and temperature during the ramming cycle.

Interpretation of these, plus other results, suggests that the Solar Turbines combined cycle system is relatively stable during the ice-ramming cycle.

Evaluation of data presented in a paper entitled "A Dynamic Analysis of a GOGAS Propulsion Plant" also suggests that the combined cycle system is able to accept the rapid manoeuvres associated with icebreaking. This analysis concluded that:

- steam turbine power does not fluctuate rapidly with changes in gas turbine power
- rapid excursions in gas turbine power are not "seen" by the steam system
- the COGAS system is a "stable" system capable of power excursions of over 15 MW (gas turbine) in seconds with an additional 5 MW (steam turbine) which takes 2 to 5 minutes to respond.

Figure 5 depicts an indication of COGAS power fluctuation during this ice ramming cycle, utilizing the data contained in the aforementioned paper. As gas turbine power decreases, available steam turbine power slowly decreases. Gas turbine power eventually increases returning to full power during the cycle but the steam turbine response is delayed as a result of its inherent thermal lag. The steam turbine eventually returns to full power well before contact with the ice edge.

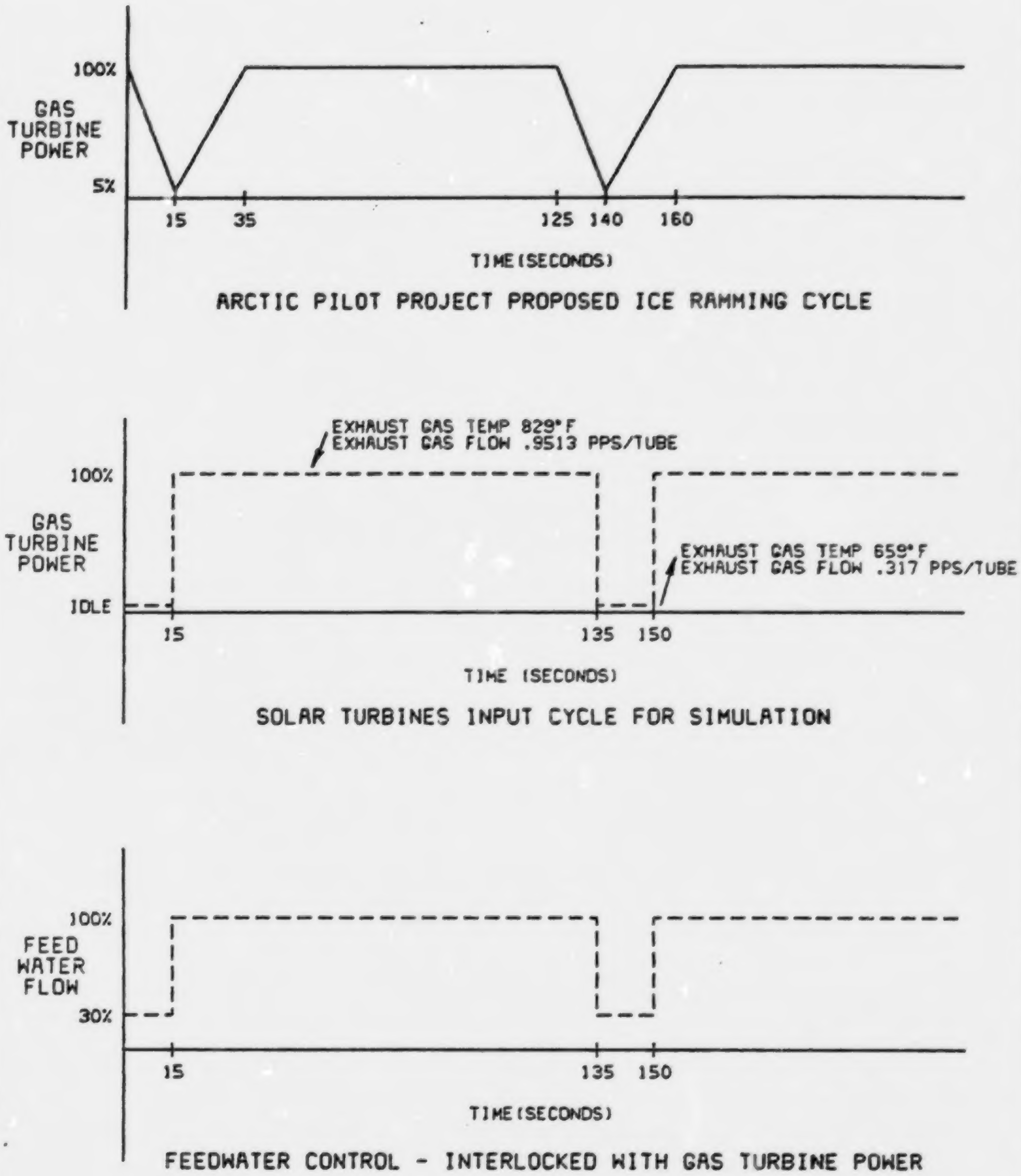

FIGURE 2
INPUT TO THE SOLAR TURBINES INTERNATIONAL SIMULATION OF ICE RAMMING CYCLE

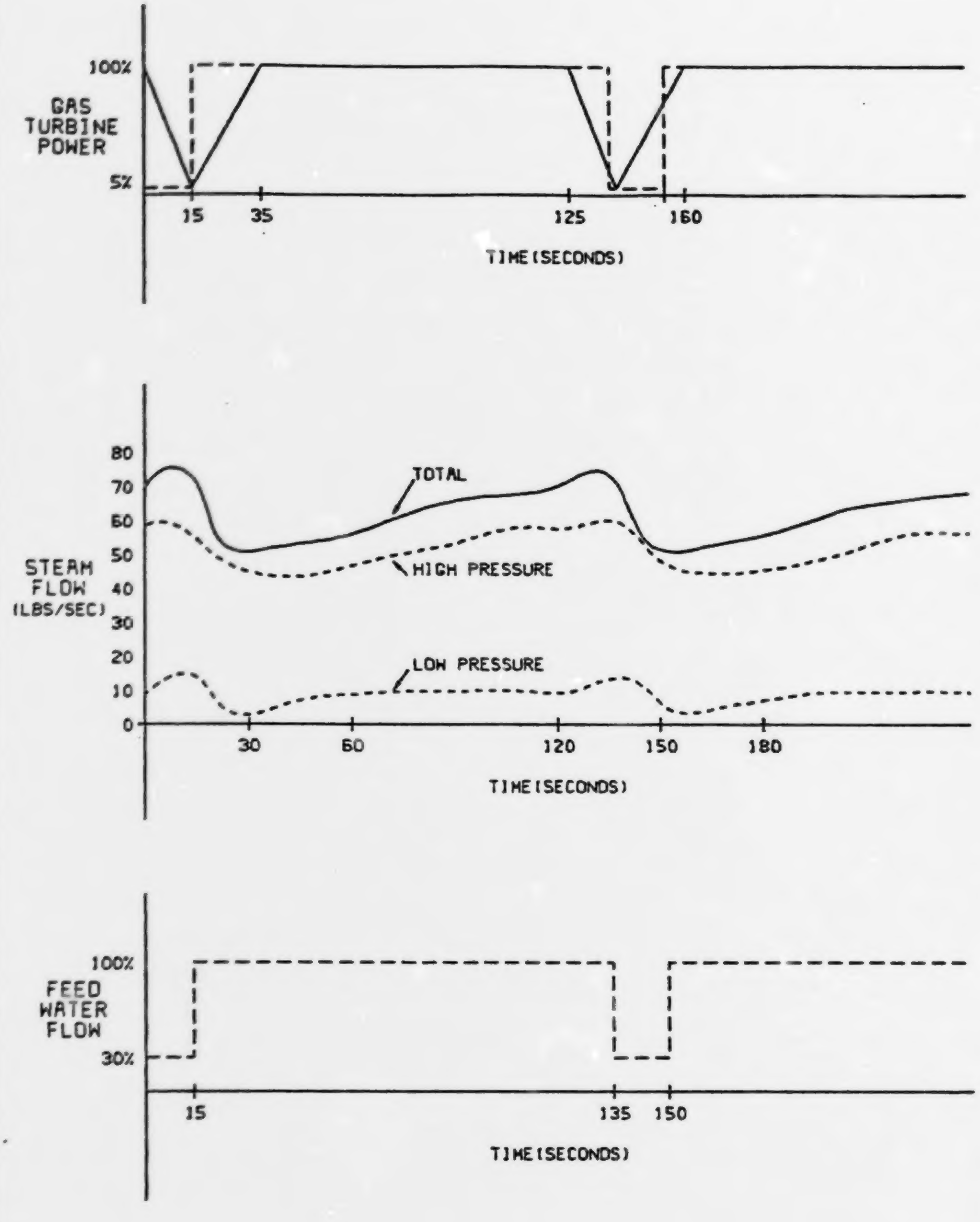

FIGURE 3
RESULTS OF SOLAR TURBINES INTERNATIONAL SIMULATION OF ICE RAMMING CYCLE

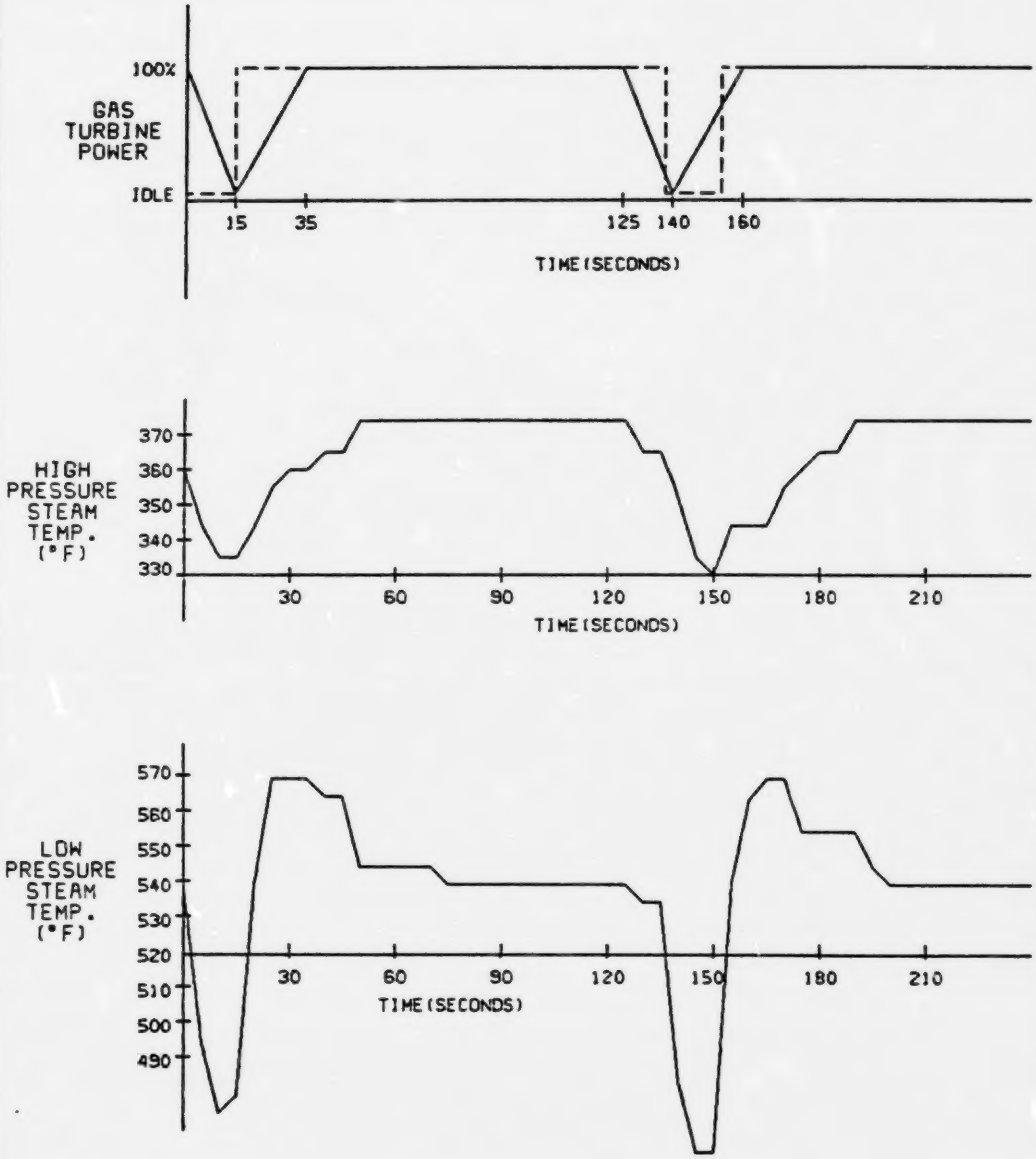

FIGURE 4
RESULTS OF SOLAR TURBINES INTERNATIONAL SIMULATION OF ICE RAMMING CYCLE

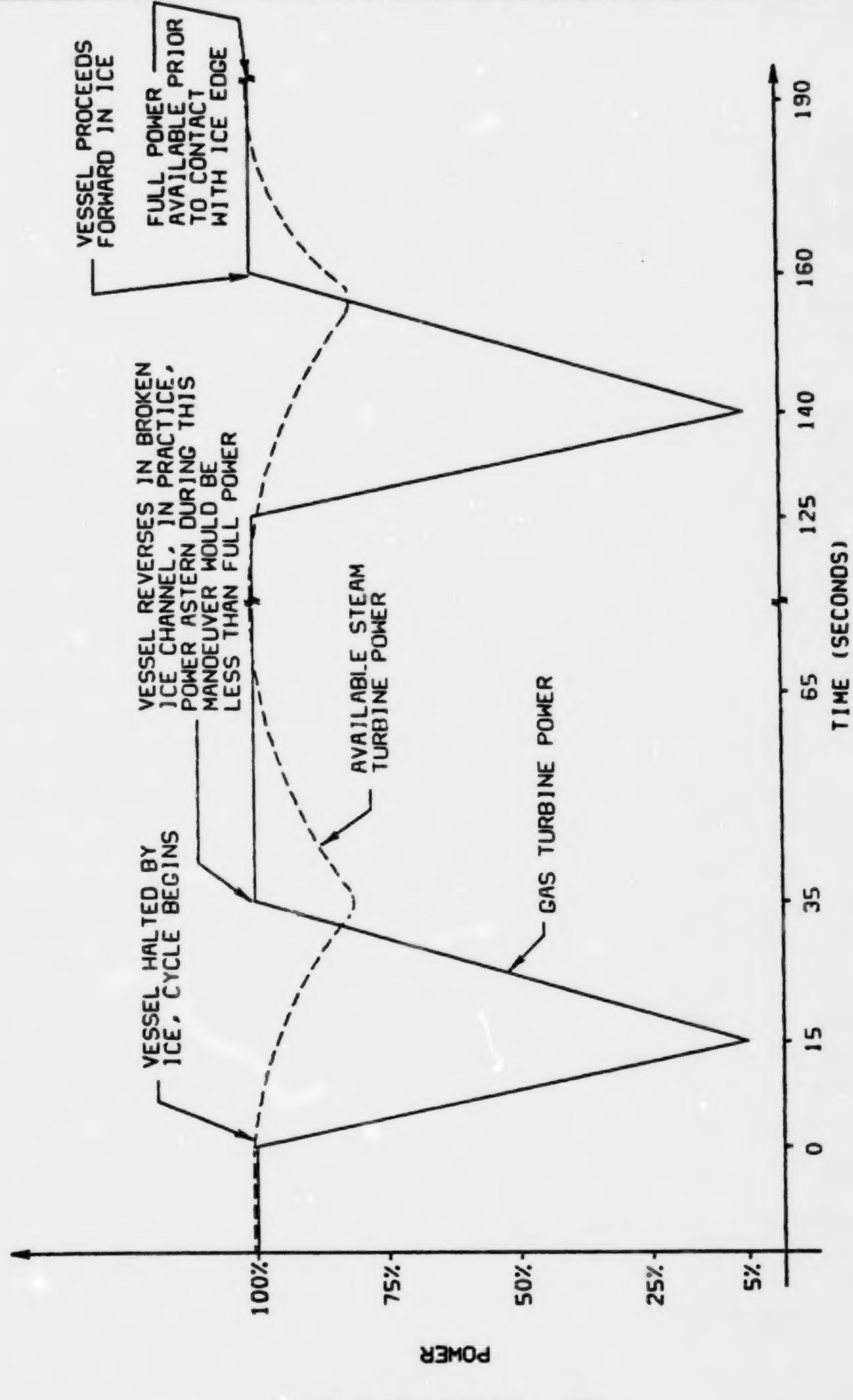

FIGURE 5
ESTIMATED FLUCTUATION
OF COGAS POWER DURING
THE ICE RAMMING CYCLE

These studies, albeit brief, do depict a tendency towards the ability of combined cycle to withstand the rigors of Arctic icebreaking.

As yet, there does not exist, with the exception of one Russian vessel, a combined cycle system operating in a marine application. There are many land-based gas turbine systems which utilize waste heat boilers to improve efficiency and there are many marine vessels are equipped with waste heat boilers to produce steam for ship service purposes. However, the lack of marine experience of the COGAS propulsion system presents uncertainty to an operator considering the installation of this plant in a Canadian Arctic icebreaker.

This study contacted manufacturers and operators of land-based combined cycle systems to ascertain their understanding of the reliability of COGAS systems. These suppliers and operators were consistent in advising that the COGAS plants with which they had been associated, had achieved excellent reliability.

This study also reviewed the incidents recorded in the Lloyd's Register of Shipping for marine vessels having:

- main membrane wall oil-fired boilers developing a minimum of 15 MW power and 4 MN/m^2 steam pressure
- steam turbines and condensers producing a minimum of 15 MW power
- vessels of Lloyd's Register class constructed between January, 1960 and December, 1979.

The results of this survey, shown in Table 1, concluded that this equipment had achieved excellent operating success in a marine application.

Although it is difficult to apply these results to a COGAS system installed in a Canadian Arctic icebreaker, these results do portend well for the operating success of this propulsion system.

This study concludes that a combined cycle propulsion system appears to offer the ability to:

- withstand the rigors of Canadian Arctic icebreaking duty
- deliver maximum power when most needed
- perform efficiently and reliably in an icebreaking application.

The combined cycle propulsion system appears to be technically suitable for installation in a Canadian Arctic icebreaking vessel.

TABLE 1
SURVEY OF BOILER, STEAM TURBINE AND CONDENSER INCIDENTS

POWER (MW)	EQUIPMENT	NUMBER OF INCIDENTS	NUMBER AT RISK	MONTHS OF SERVICE	INCIDENCE RATE	REMARKS
15 - 22.5	Steam Turbine	88	236	11,216	1/10.6 yrs	Most defects occur in bearings, valves, blading
15 - 22.5	Condensers	17	236	11,216	1/55.0 yrs	Most downtime results from tube failure
15 - 22.5	Boilers	22	25	1,166	1/4.4 yrs	Tube failures constitute greatest incidents
22.5 - 30	Steam Turbines	82	338	13,053	1/13.2 yrs	Most defects occur to bearings, blading
22.5 - 30	Condensers	13	338	13,053	1/83.7 yrs	Most defects occur to tubing
22.5 - 30	Boilers	53	149	5,143	1/8.1 yrs	Most defects occur to tubing and valves
30 - Plus	Steam Turbines	43	84	3,188	1/8.3 yrs	Most defects occur to blading
30 - Plus	Condensers	2	84	3,188	1/132.8 yrs	Most defects occur to tubing
30 - Plus	Boilers	0	28	832	-	-

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By

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Calgary, Alberta

For

Arctic Pilot Project, Inc.
Calgary, Alberta

and

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This report supplements the initial studies conducted by Pratt and Whitney by assessing the technical feasibility of installing a combined gas turbine and steam turbine propulsion system in a Canadian Arctic icebreaker. This study addresses, in particular, the ability of a combined cycle propulsion system to:

- withstand rigorous icebreaking duties
- deliver maximum power when required
- perform reliably in an icebreaking application

SUMMARY

This study presents a technical evaluation of a combined gas turbine and steam turbine propulsion system for Canadian Arctic icebreaking vessels.

The principle objectives of the study were to determine the ability of the combined cycle propulsion system to:

- withstand rigorous icebreaking duties
- deliver maximum power when required
- perform reliably in an icebreaking application

The study addresses these issues through the evaluation of computer-based simulation of the combined cycle propulsion system conducting icebreaking manoeuvers. These simulations suggest that the combined cycle propulsion system is able to withstand the rigors of icebreaking duties and is able to deliver maximum power when most needed.

The study addresses the reliability of the combined cycle propulsion system through evaluation of the experience of:

- land-based combined cycle plants
- marine waste-heat boilers
- marine high-pressure boilers and steam turbines

Review of this data suggests that a combined cycle propulsion system should perform reliably in a Canadian Arctic icebreaking vessel.

The study concludes that a combined cycle propulsion system appears to be technically suitable for installation in a Canadian Arctic icebreaking vessel.

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1. INTRODUCTION

In early 1974, the influence of the Organization of Petroleum Exporting Countries (OPEC) cartel was first seen as the price of the benchmark crude oil, Saudi Arabian light, climbed almost 300% to U.S. $9.60 per barrel. Since this auspicious beginning, OPEC has continued to exert its strength. In the two years between January 1979 and January 1981, the contract price of Saudi Arabian light crude oil was increased eight times as the per barrel value climbed 240% from U.S. $13.34 to U.S. $32.00. In October 1981, the contract price of Saudi Arabian light crude oil was increased to U.S. $34.00 per barrel where it remains in late 1982. Figure 1 depicts this movement in greater detail.

This dramatic increase in oil prices has had a pronounced influence upon many industries and particularly marine operations where substitutes for liquid fuel are not always practical. It was not long ago when ship operators might have considered fuel costs as a minor expense of their business. The substantial increase in fuel price has changed this attitude. Ship designers now strive for propulsion system efficiency as a means to reducing their burgeoning fuel expenses.

As fuel prices climb, it becomes economical to install equipment which captures energy that otherwise might be lost. Whereas, a few years previous, the capital cost of this equipment was not economically justified, the savings now available with current fuel prices suggests this equipment be installed. For instance, in systems where a large quantity of heat is rejected in exhaust gases, such as diesel engines and gas turbines, system efficiency can be improved by installing waste heat boilers to generate steam. On a marine vessel, this steam may be used to supplement the power available from the prime mover or alternatively to generate electrical power for vessel auxiliaries through steam turbines. In the case of a diesel engine, which has a comparatively flat fuel characteristic, approximately 10% extra power can be readily recovered by means of a waste heat boiler system which would result in a corresponding decrease in the system's overall specific fuel consumption. In the case of a gas turbine, the potential for heat recovery, due to the large air mass flow and high exhaust temperatures is as high as 25% of the gas turbine power at full load, rising to as much as 40% at low loads. Hence, by the use of relatively simple and proven marine machinery, the system's overall thermal efficiency can be increased substantially to approximately 43/44% at full power.

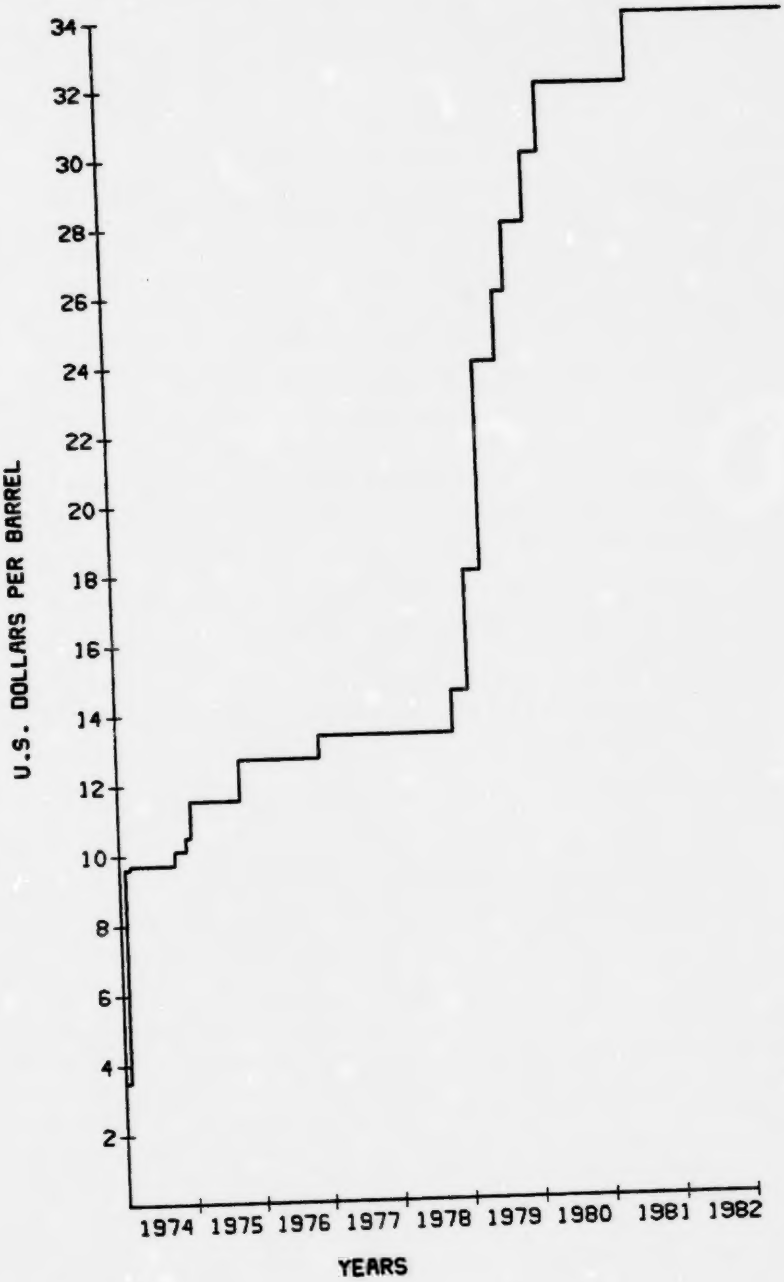


FIGURE 1
PRICE OF SAUDI ARABIAN
LIGHT CRUDE OIL
1974 - 1982

In 1978 and 1979, Pratt and Whitney Canada Ltd. conducted a study for the Department of Transport, which investigated the feasibility of installing in a marine application a propulsion system consisting of aero-derived gas turbines, waste heat boilers and steam turbines. This study concluded that it is feasible to install a combined gas turbine and steam turbine propulsion system in a marine application.

This report will supplement the initial studies conducted by Pratt and Whitney by assessing the technical feasibility of installing a combined gas turbine and steam turbine propulsion system in a Canadian Arctic icebreaker, specifically the liquid natural gas carriers, proposed by the Arctic Pilot Project and being designed to Class 7 of the Arctic Shipping Pollution Prevention Regulations.

This report will assess:

-the design of a combined gas turbine and steam turbine propulsion system appropriate for the icebreaking liquid natural gas carriers

-a possible method of control for the combined gas turbine and steam turbine system

-the ability of the combined gas turbine and steam turbine system to withstand the rigorous duty of an icebreaker

-the reliability of combined gas turbine and steam turbine system

-the research and development being conducted which might affect future combined gas turbine and steam turbine systems.

Combined gas turbine and steam turbine propulsion offers substantial fuel savings as previously described. However, for an icebreaker, the potential benefits are even greater. For instance, the heat or energy that is usually rejected in the condenser of the steam system may be utilized to supply heat to accommodation areas, ballast tanks or any other areas on the vessel that require relief from cold temperature. In shore-based installations, where virtually all this waste heat may be utilized, the efficiency of the combined gas turbine and steam turbine system may be as high as 80% with the only losses resulting from radiant heat leakage, mechanical losses, plus certain residual losses in the gas turbine exhaust and steam turbine condenser. For the icebreaking LNG carriers, it is estimated that the thermal efficiency of a combined gas turbine and steam turbine plant may be as high as 60% if the residual energy in the steam at the condenser cooling coils is utilized for vessel heating in Arctic conditions.

Although waste heat recovery using boilers situated in the exhaust gas flow of diesel engines and gas turbines, with the steam generated being used to drive steam turbine alternator sets, has been common practice for many years in ocean going vessels, a similar waste heat recovery system fitted to supplement prime mover power has not yet found an application in a vessel built outside the Soviet Union. Although the various individual components of the combined gas turbine and steam turbine propulsion plant have significant marine experience, it is the lack of marine history of the total plant which presents uncertainty to an operator considering its installation, particularly for an Arctic icebreaker. This study will identify potential difficulties of a combined gas turbine and steam turbine plant installed on an Arctic icebreaker and draw conclusion regarding the technical feasibility of the system in this application.

2. DESIGN OF A COMBINED GAS TURBINE AND STEAM TURBINE PROPULSION SYSTEM

The propulsion system evaluated in this report consists of aero-derived gas turbines equipped with waste heat boilers in the exhaust stack to produce steam for additional propulsive power. This system has a number of designators including:

-COGAS - referring to Combined Gas and Steam Turbine

-combined cycle - referring to the Brayton cycle of the gas turbine and the Rankine cycle of the steam turbine

-RACER - a United States' Navy designator referring to Rankine Cycle (steam turbine) Energy Recovery

The gas turbine and steam turbine system evaluated in this report has a power appropriate for a Canadian Arctic icebreaker and specifically the Class 7 liquid natural gas carriers proposed by the Arctic Pilot Project (APP). Preliminary design of these vessels envisaged 150 megawatts installed power distributed through electric transmission to three propeller shafts. Three types of aero-derived gas turbines have been considered for this application: the Pratt and Whitney FT-4C, the Rolls Royce RB 211 and the General Electric LM 2500. The latter two turbines are considered "second generation" machines in that these are characterized by higher air compression ratios and firing temperatures and therefore have a lower specific fuel consumption than the "first generation" FT-4C. As a result, the RB 211 and LM 2500 are preferred for an Arctic icebreaker and the COGAS system developed in this report will be predicated upon these two gas turbines.

With the electric transmission system proposed for the LNG carriers, the gas turbine prime movers operate at constant speed, driving alternating current generators. These generators supply power at a fixed frequency to a converter which provides a variable frequency output. The frequency converter is comprised of thyristors which essentially block the current flow until receiving a pulsed signal which initiates passage of current. In this way, the fixed frequency input can be modified to a selected frequency output by adjustment to the time frame of pulses. The output from the thyristors feeds synchronous motors on the propeller shafts. Propeller speed is controlled by the pulse frequency, and propeller torque by the amplitude of the motor current. Figure 2 describes schematically this propulsion system.

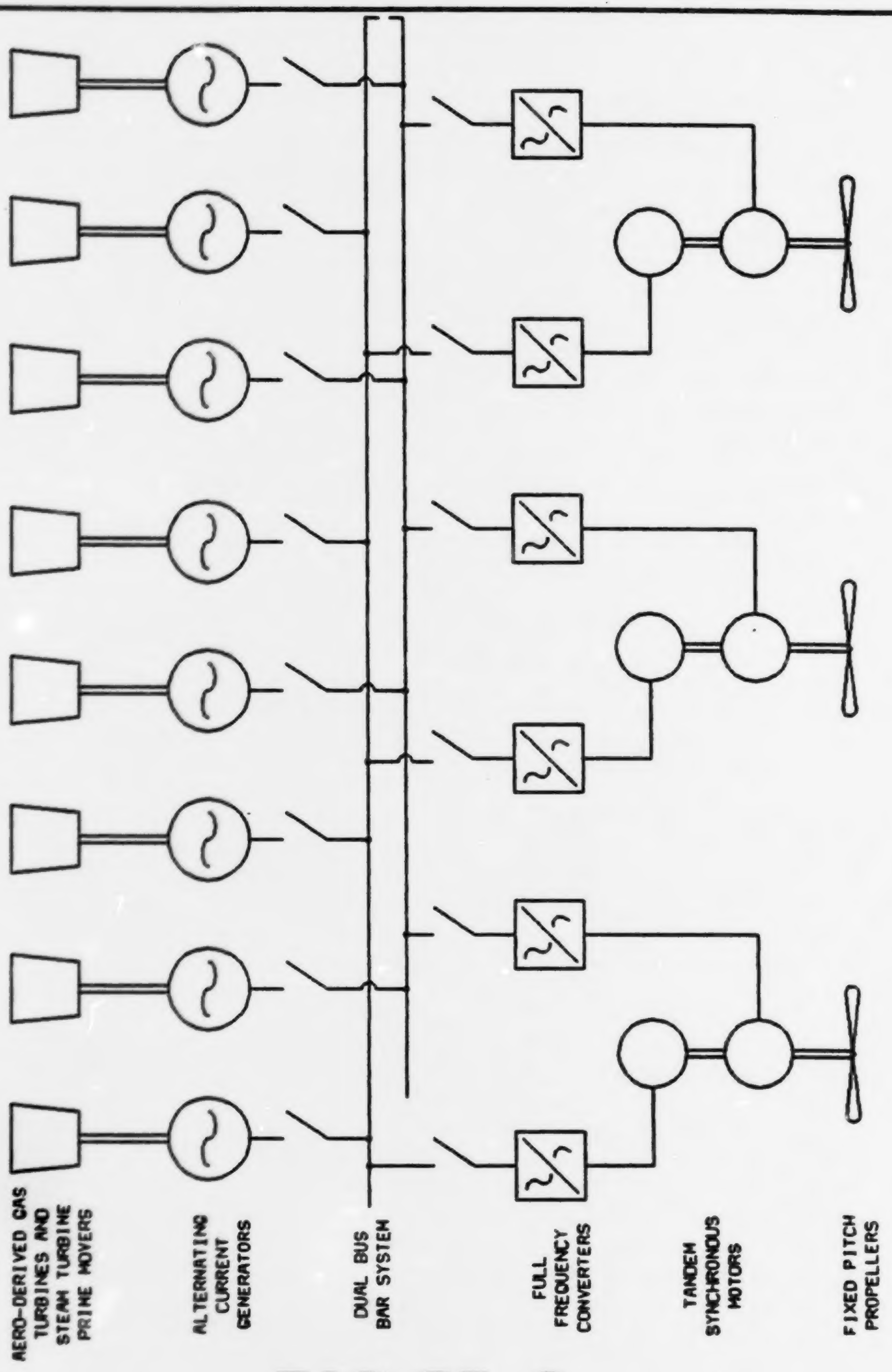

AERO-DERIVED GAS TURBINES AND STEAM TURBINE PRIME MOVERS
ALTERNATING CURRENT GENERATORS
DUAL BUS BAR SYSTEM
FULL FREQUENCY CONVERTERS
TANDEM SYNCHRONOUS MOTORS
FIXED PITCH PROPELLERS

FIGURE 2
POSSIBLE PROPULSION SYSTEM
FOR ICEBREAKING LNG CARRIERS

The Arctic Pilot Project is considering an alternative propulsion system which is technically feasible and may offer capital cost benefits and improved transmission efficiency. This system consists of similar prime movers connected through a mechanical gearbox and shafting to controllable pitch propellers which are protected from ice masses by nozzles. Although the technical evaluations which follow are predicated upon an electric transmission system, the conclusions are also relevant for the mechanical transmission/controllable pitch propellers system.

The Arctic Pilot Project vessels are being designed to consume natural gas fuel to the greatest extent practicable. Natural gas fuel is beneficial to a COGAS system for many reasons including:

-higher allowable firing temperature of the gas turbine allowing the COGAS system to produce more power

-gas turbine exhaust gas contains few contaminants which might initiate waste heat boiler corrosion.

The gas turbines will consume occasionally a distillate fuel which will contain more impurities than the natural gas. The COGAS system must be designed to consume reliably and effectively, all grades of fuel under consideration.

2.1 Description of the COGAS System

The description of the COGAS system which follows is predicated upon the power required by the Arctic LNG carriers of approximately 150 megawatts equally distributed to three propeller shafts. This power is provided by gas turbines with waste heat boilers in the exhaust stacks producing steam for additional propulsive power.

2.1.1 Aero-derived Gas Turbine

The aero-derived gas turbine is a light weight machine which provides a favourable power to weight ratio. This light weight construction allows the gas turbine to more readily accept rapid changes in load without incurring damage from thermally induced stresses. This ability to accept rapid load variations by means of a simple control system renders the gas turbine a suitable prime mover for application in icebreakers.

The output power from gas turbines is influenced by the ambient air temperature. In cold climates, the potential power from gas turbines increases due to lower inlet air combustion temperatures and higher air density, and the ability to burn more fuel without degradation of the hot section of the machine. Practically, however, the power is limited by the stresses in the compressor section of the machine which occur with the increase in air density. In high ambient temperatures, power is limited by the hot section temperatures and the ability of the material to carry the operating stresses at these temperatures for prolonged periods without premature failure. Hence the gas turbine has the desirable characteristic for an icebreaking vessel in that the maximum powers from the engine are availabe at the low ambient temperatures.

In the case of a COGAS system, the steam output from the system is dependent upon the temperature and mass flow of the gas turbines exhaust gases. As the ambient air temperature decreases, the mass flow of the gases increases to a maximum dependent upon the limitation in the gas turbine compressor due to stress. However, the exhaust gas temperature also decreases. The result is that the steam production, superheat temperature and hence steam turbine power tends to decrease with ambient air temperature. The net effect is that the steam generated in -40°C ambient is approximately 60% of that generated at ISO (15°C). The gas turbine power is increased by approximately 20% at -40°C relative to ISO resulting in an overall increase in power of 6%. Although the increase in system output is not large, the trend is favourable to the system operating in cold ambient conditions.

The COGAS system consisting of two LM 2500 or RB 211 gas turbines with waste heat producing steam for additional propulsive power provides approximately 50 and 56.5 megawatts at 15°C respectively. Figure 3 depicts a schematic diagram of this propulsion system.

Figures 4 through 7 depict the manufacturer's estimated operating characteristics of the LM 2500 and the RB 211. Figures 8 and 9 depict the estimated power and fuel consumption rates available from the LM 2500 and RB 211 COGAS systems.

2.1.2 Waste Heat Boiler

The design of the waste heat boiler will be a compromise between efficiency in extracting heat from the gas turbine exhaust and cost effectiveness. The following paragraphs describe a waste heat boiler design suitable for Arctic operation.

2.1.2.1 Approach Temperature

The approach temperature represents the difference between exhaust gas entry temperature to the superheater and steam temperature leaving the superheater. The lower the approach temperature and accordingly, the higher the superheated steam temperature, the greater the steam cycle efficiency. However, the lower the approach temperature, the greater are boiler capital costs. An approach temperature of 44°C (80°F) is utilized within the "boiler industry" with minor variation between individual manufacturers.

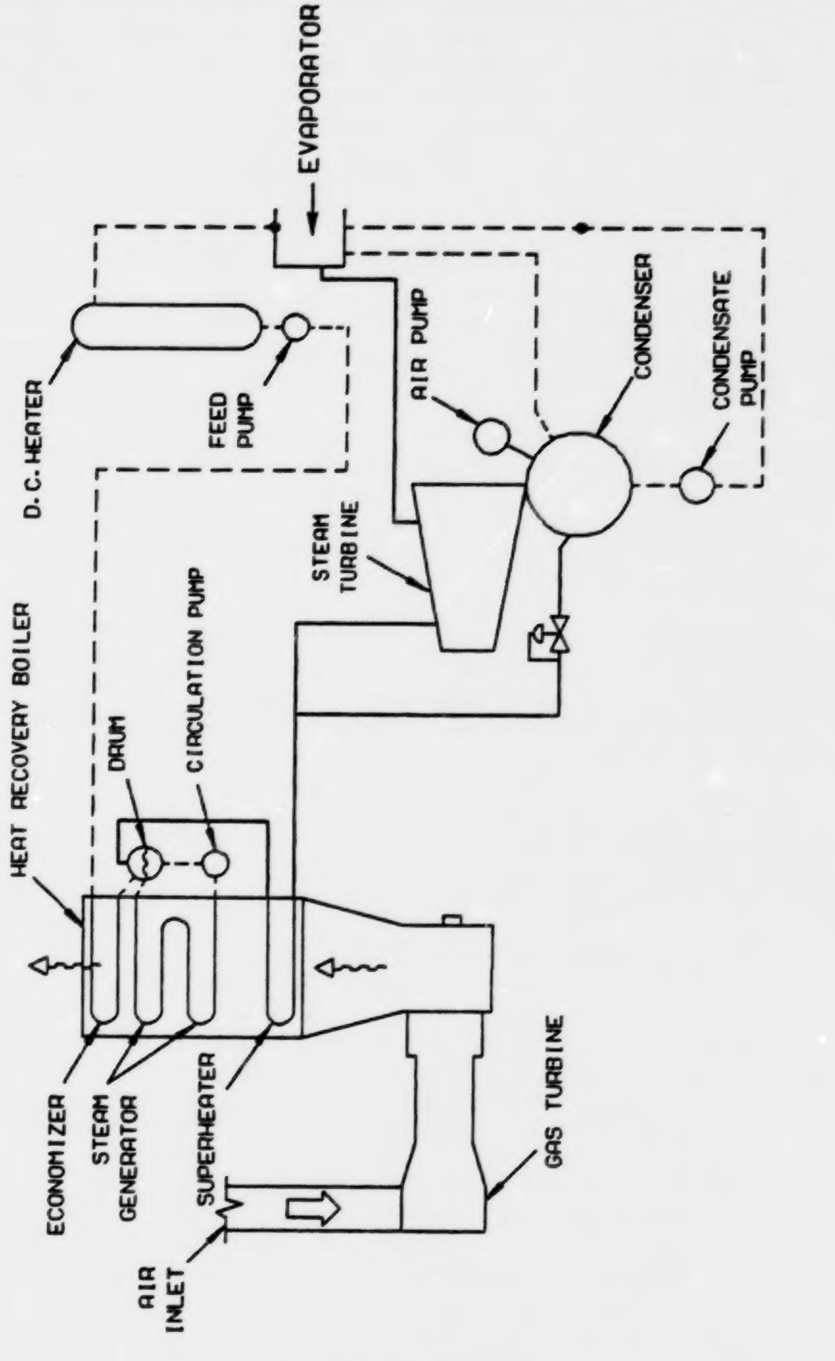
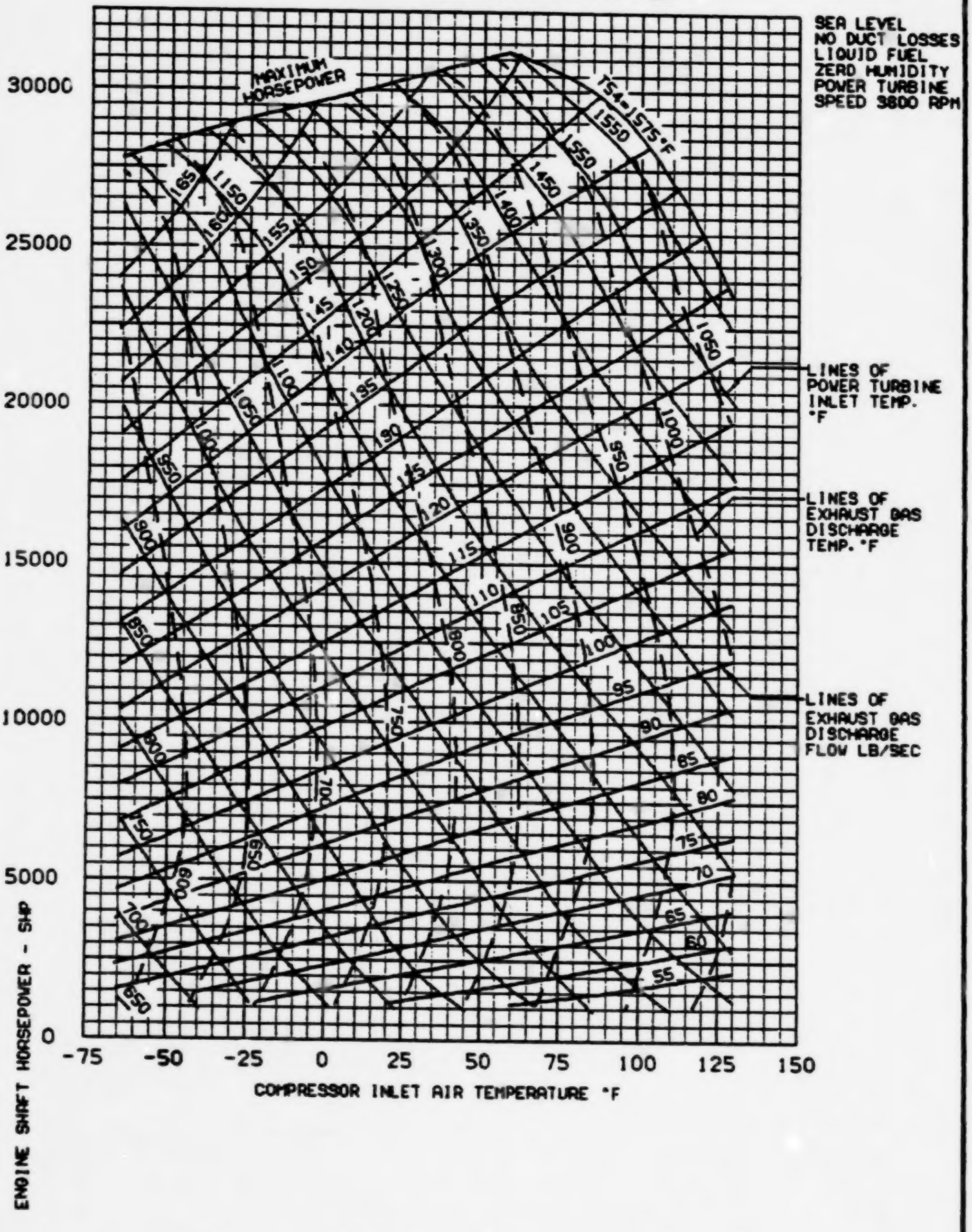

EVAPORATOR
D.C. HEATER
FEED PUMP
AIR PUMP
CONDENSER
CONDENSATE PUMP
STEAM TURBINE
HEAT RECOVERY BOILER
DRUM
CIRCULATION PUMP
ECONOMIZER
STEAM GENERATOR
SUPERHEATER
GAS TURBINE
AIR INLET

FIGURE 3
COGAS PROPULSION SYSTEM



SOURCE: LM2500-30 MARINE GAS TURBINE PERFORMANCE
DATA: GENERAL ELECTRIC JULY 1980

FIGURE 4
LM 2500 GAS TURBINE
ESTIMATED AVERAGE
PERFORMANCE

SOURCE: LM2500-30 MARINE GAS TURBINE PERFORMANCE
DATA: GENERAL ELECTRIC JULY 1980

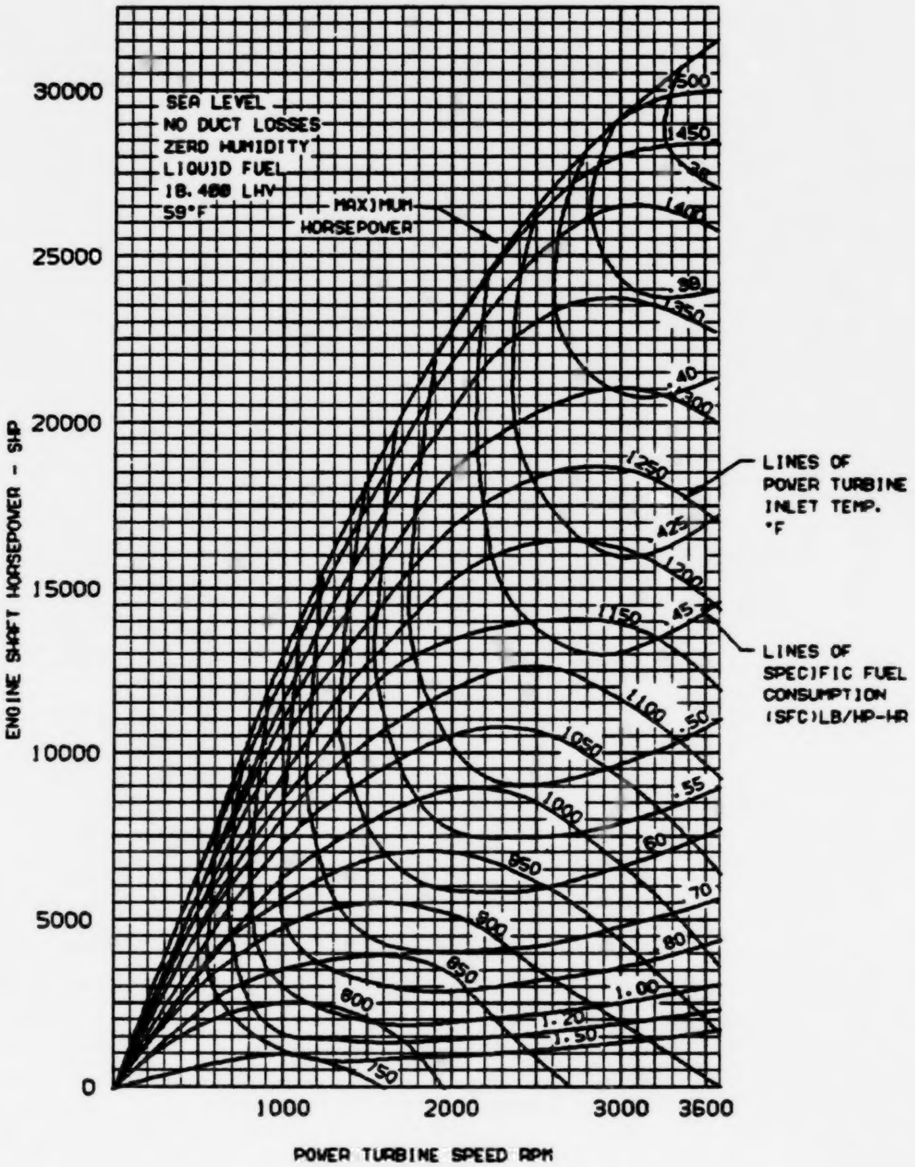


FIGURE 5
LM 2500 MARINE GAS TURBINE
ESTIMATED AVERAGE
PERFORMANCE

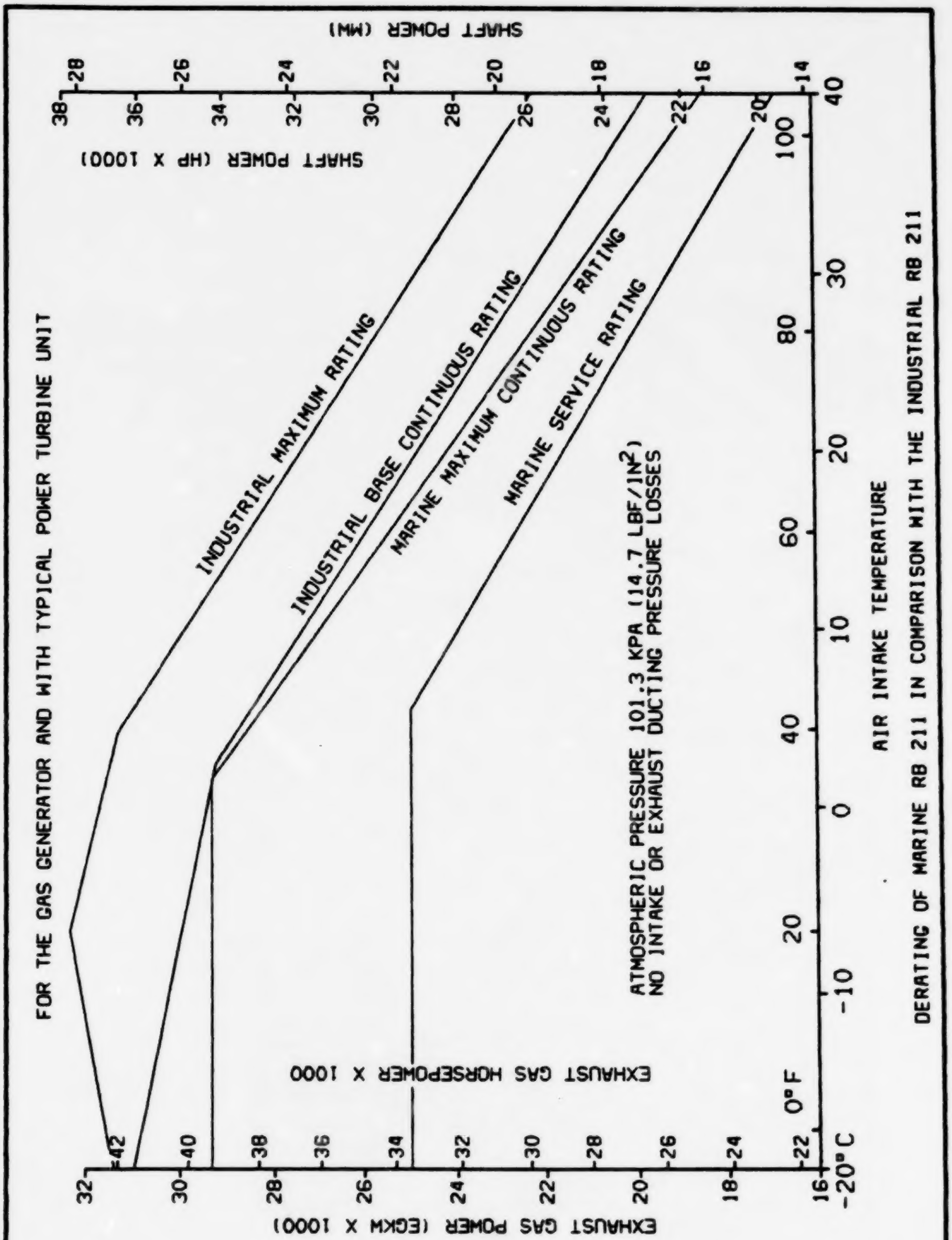


FIGURE 6
ESTIMATED PERFORMANCE OF
THE ROLLS ROYCE RB 211
GAS TURBINE

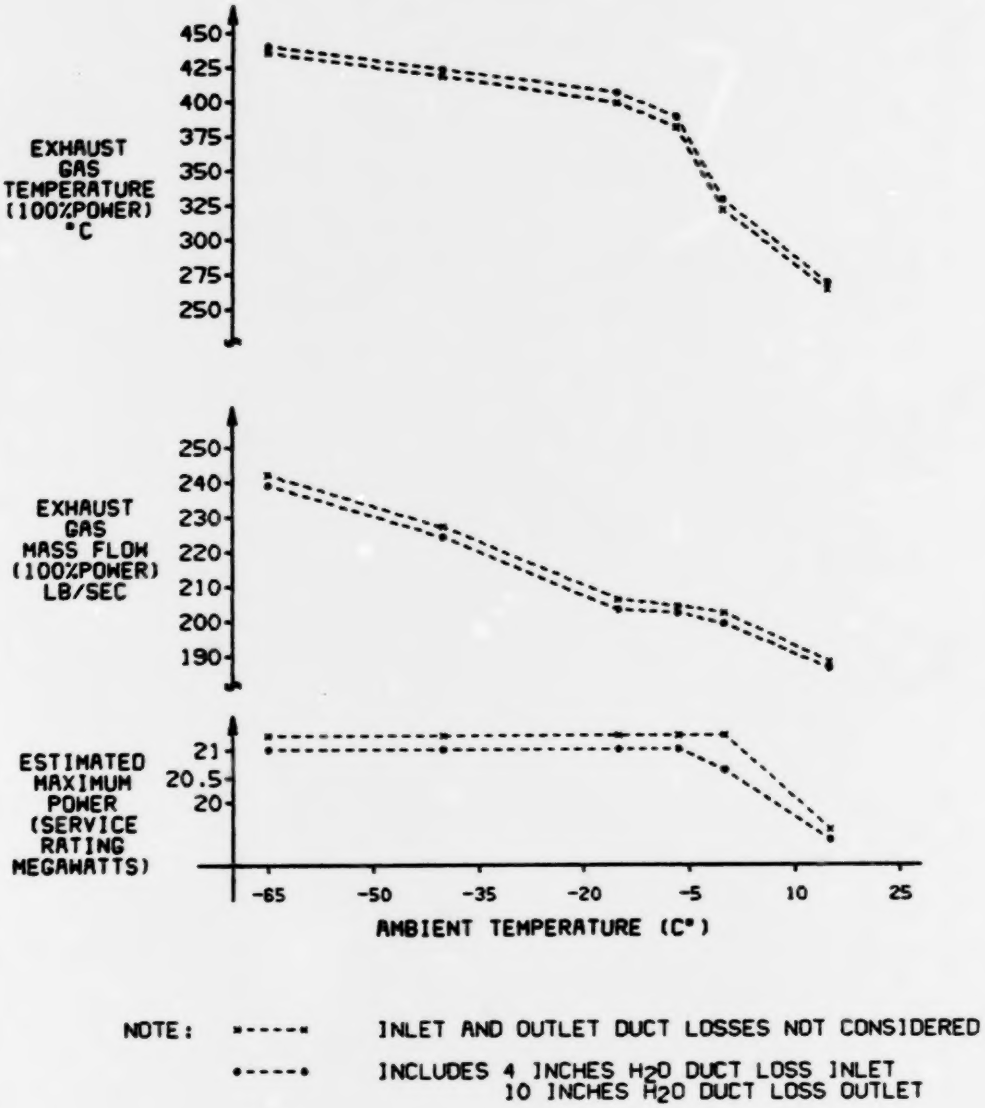


FIGURE 7
ESTIMATED OPERATING PERFORMANCE
ROLLS ROYCE RB 211

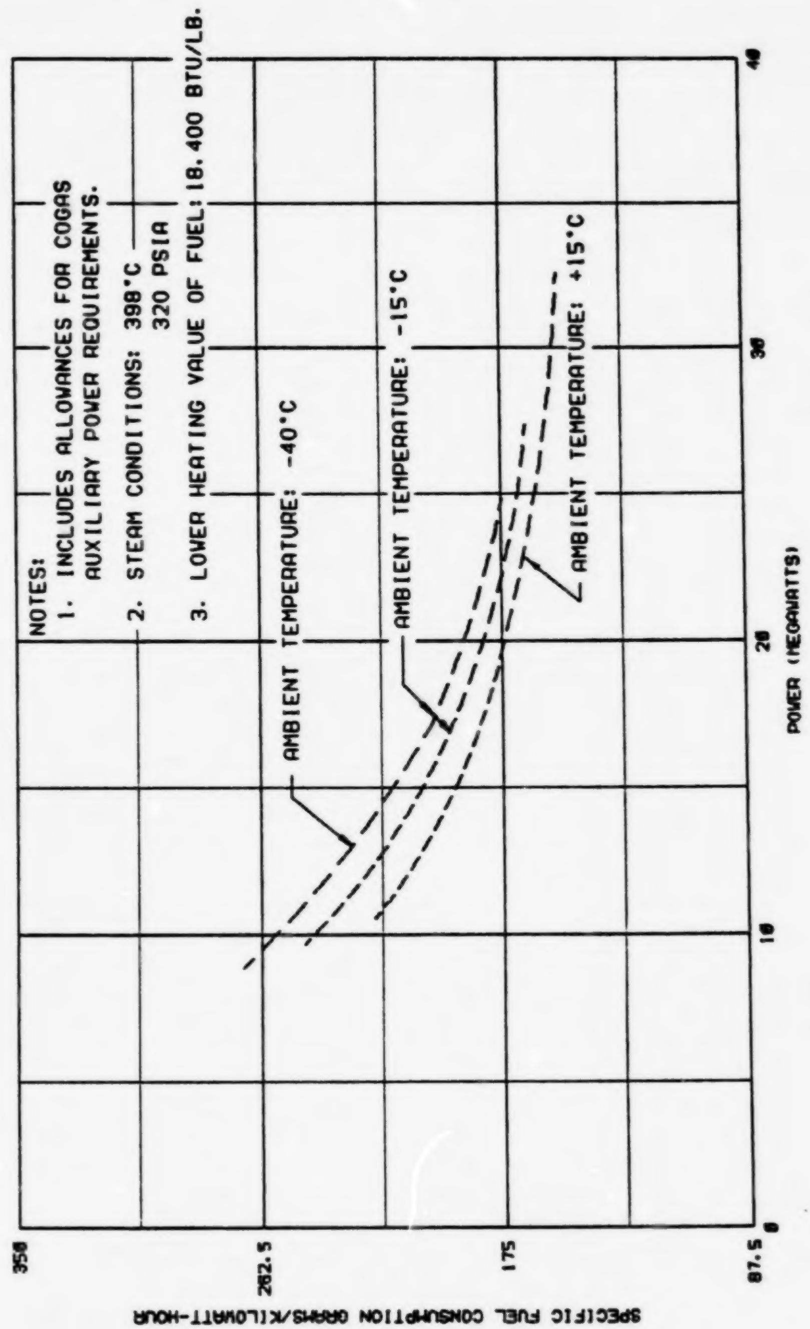
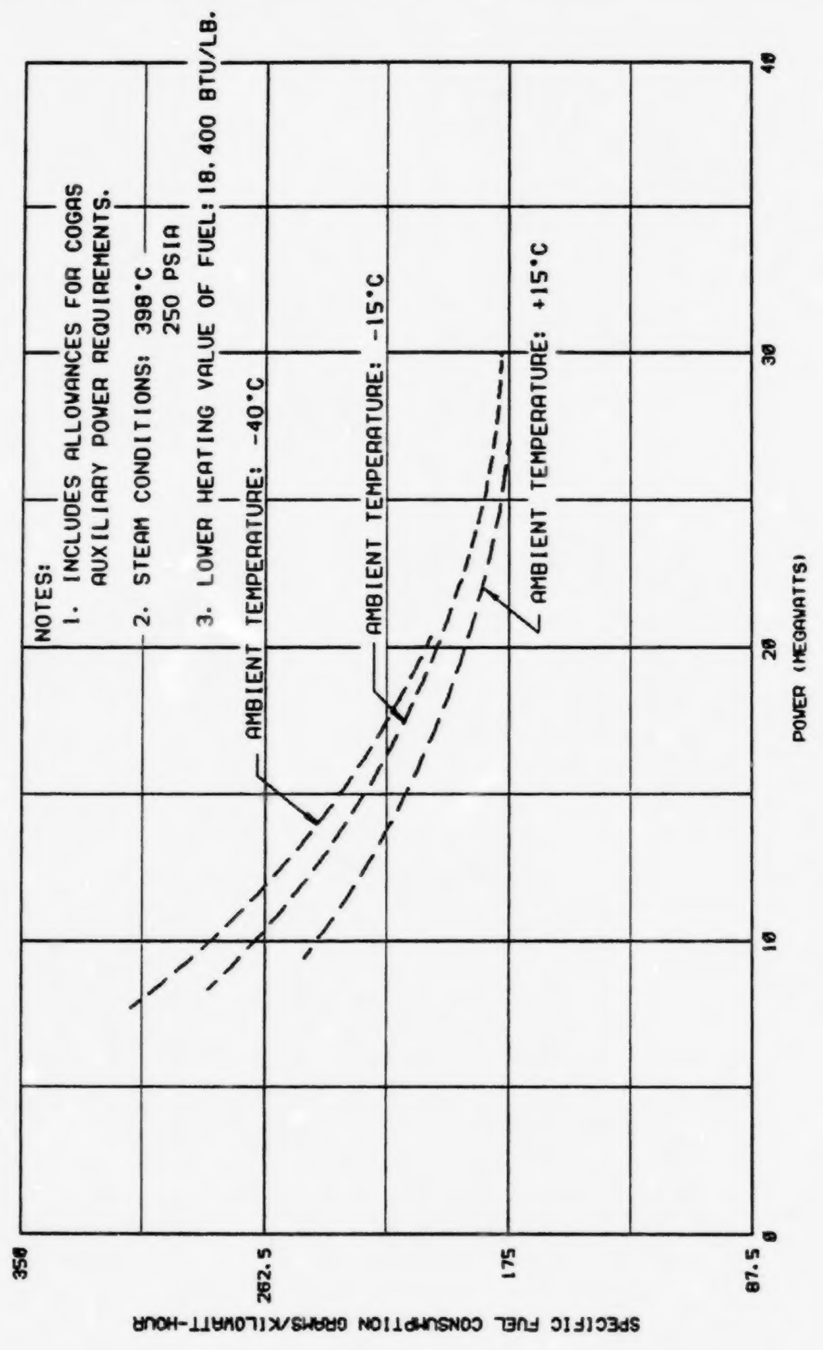


FIGURE 8
ESTIMATED LM 2500
COGAS FUEL
CONSUMPTION RATES



NOTES:

1. INCLUDES ALLOWANCES FOR COGAS AUXILIARY POWER REQUIREMENTS.
2. STEAM CONDITIONS: 398°C
   250 PSIA
3. LOWER HEATING VALUE OF FUEL: 18,400 BTU/LB.

FIGURE 9
ESTIMATED RB 211
COGAS FUEL
CONSUMPTION RATES

2.1.2.2 Pinch Point

The pinch point represents the temperature difference between the exhaust gas leaving the boiler evaporator and the saturation temperature of the steam. The lower the pinch point, the greater the steam flow and waste heat recovery. However, practical limitations restrict the minimum pinch point. The boiler might be designed with a pinch point of 22°C (40°F). Discussions with boiler manufacturers confirmed their acceptance of this value. A pinch point temperature of 22°C is considered an appropriate compromise for an icebreaker between efficiency and capital cost.

2.1.2.3 Exhaust Gas Stack Temperature

The APP carriers are considering consuming natural gas fuel to the greatest extent practicable. The natural gas carried in liquid form by the carriers is a very "sweet" gas having negligible sulphur content. As a result, with the gas turbines operating on this natural gas fuel, the temperature of the exhaust gas at the outlet of the boiler may be reduced to low levels to improve system efficiency without incurring corrosion of boiler tubing as a result of sulphuric acid attack.

However, to ensure flexibility of operation, the boiler must also be designed for liquid fuel consumption by the gas turbine. The acceptable exhaust stack temperature is a function of the dew point of the exhaust gases and the sulphur content of the fuel which, in turn, is identified by the fuel specification. The aero-derived gas turbines require a distillate fuel which is usually specified with a low sulphur content. Table 1 depicts certain specifications for a marine diesel oil suitable for consumption by an aero-derived gas turbine. Figure 10 presents the dew point temperature of the exhaust gases as a function of the sulphur content of the fuel. To avoid the sulphur dew point, and possible corrosion of boiler tubing, a minimum exhaust gas stack temperature of 126.7°C (260°F) is necessary.

2.1.2.4 Backpressure Created by Waste Heat Boiler

Resistance to the air flow entry and exhaust from the gas turbine will create a loss of prime mover power and efficiency. As a result, inlet and exhaust ducts are installed as straight as possible. Also, the waste heat boiler should be designed to minimize back pressure to gas flow. However, the boiler must be efficient which will necessitate a compromise between backpressure on the gas turbine exhaust and boiler size and heat transfer efficiency.

General Electric, in their description of LM 2500 operating characteristics, recommend that "the backpressure must not exceed 20 inches H_2O (gage) total measured at the exhaust collector exit".

The conceptual designs of exhaust gas boilers received from manufacturers in the course of this study provide total pressure drop, for boiler and ducting, of between 10 inches and 12 inches H_2O (gage) which appears to be an appropriate compromise between boiler efficiency and gas turbine inefficiency created by exhaust stack backpressure.

TABLE 1
TYPICAL FUEL SPECIFICATION FOR AERO-DERIVED GAS TURBINES

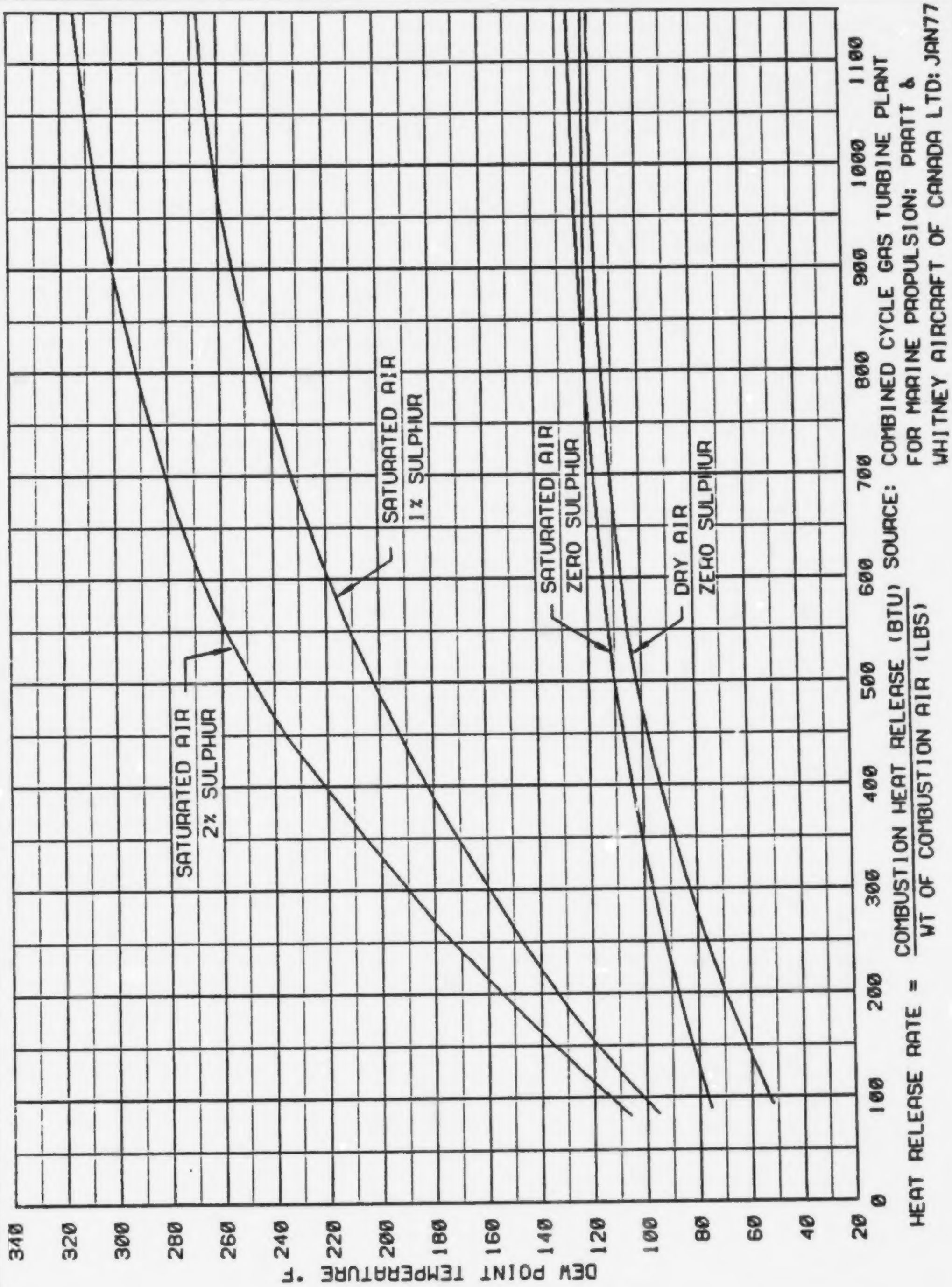
Fuels meeting the following specifications are generally acceptable provided contaminants and viscosity are within tolerable limits.

Specification	Title
MIL-T-5624	Grades JP4, JP5 (NATO F-40, F-44)
ASTM D975-74	Diesel Fuel Oil, Grades 1-D and 2-D
ASTM D1655-75	Turbine Fuels
MIL-F-16884	Fuel Oil - Diesel Marine (NATO F-75, F-76)
VV-F-800A	Fuel Oil - Diesel, Grades DF-A, DF-1 and DF-2 (NATO F-54)
ASTM D2880-76	Gas Turbine Fuel Oils, Grades No. O-GT, No. 1-GT, No. 2-GT, No. 3-GT
ASTM D396-76	Grades 1, 2, and 4

Additionally, fuels should also generally satisfy the following constraints:

Viscosity, centistokes	0.5 -1.1
Ash, maximum percent	0.01
Sulfur, maximum percent	1.3
Vanadium, maximum ppm	0.5
Sodium + Potassium, maximum ppm	1.0
Lead, maximum ppm	1.0
Calcium, maximum ppm	2.0
Hydrogen Content, minimum percent	12.0
Demulsification, minutes, maximum	20.0
Carbon Residue, Maximum (100% sample)	1.0

SOURCE: Planning Document, LM 2500 Dual Fuel Burning Gas Turbines for LNG Carriers; General Electric Company; April 1978.



SOURCE: COMBINED CYCLE GAS TURBINE PLANT FOR MARINE PROPULSION: PRATT & WHITNEY AIRCRAFT OF CANADA LTD: JAN77

FIGURE 10
DEW POINT OF
SULFUROUS FUELS

2.1.2.5 Steam Conditions

The selection of steam pressure and temperature is governed by the temperature available from the exhaust gases at the selected design point of the system. The superheat steam temperature is a function of the turbine exhaust temperature and the selected approach temperature. Gas turbine exhaust temperature varies with ambient air temperature and load. Thus, a system designed for the theoretically highest superheat temperature will experience certain fluctuation in steam conditions at the superheater outlet when the ambient air temperature and load changes. The APP vessels will operate in air temperatures ranging from -55°C in the Arctic regions to 38°C in more southerly climates. Ambient temperature changes and the fluctuating gas turbine exhaust characteristics necessitates a compromise in the selection of boiler temperature and pressure. If too high a boiler pressure is selected, then a low quality steam would be produced at the lower ambient air conditions, which would introduce problems of erosion in the steam turbine. If too low a pressure is selected, the larger specific volume of steam would necessitate increased size of pipework and turbines. In the case of superheat temperature, the highest superheat temperature practically attainable would require the use of high temperature resistant alloy steels, whereas a low superheat temperature will result in loss of steam plant efficiency. The design philosophy to be adopted for an Arctic icebreaker should allow the waste heat steam system to be designed to meet these divergent requirements while maintaining the existing "State of the Art" in marine technology. The waste heat boiler pressure and steam temperature should be selected to provide an acceptable degree of superheat to the steam while operating the gas turbines at the lowest ambient temperatures and therefore exhaust gas temperatures. The lowest acceptable superheated steam temperature will then be governed by the degree of wetness of the steam at the exhaust end of the steam turbine, although this can be controlled to some extent by regulation of the condenser vacuum.

For a Canadian Arctic icebreaker, superheated steam conditions of approximately 400°C (752°F) temperature and 2.53 MN/m^2 (25 atmospheres) pressure seem an appropriate selection. However, the actual selection will depend upon the exhaust temperature of the gas turbine for the "design" ambient temperature.

2.1.2.6 Dual Pressure Boiler

Efficiency of the waste heat boiler system can be increased by the adoption of a dual pressure boiler. Dual pressure boilers are, in effect, two boilers situated in series in the exhaust gas stream, each boiler being designed for a particular temperature and pressure to extract the maximum possible heat from the exhaust.

The use of a dual-pressure system improves waste heat boiler efficiency approximately 12% relative to a single pressure system at the exhaust gas temperatures being considered. The steam portion of the COGAS system provides 9% to 10% increase in total system efficiency so the installation of a dual pressure boiler would provide an overall 1% benefit to total system efficiency.

The dual pressure system is typically adopted in land installations for base load plants and may be suitable for ocean going vessels on long sea voyages which operate essentially under steady load. A dual pressure plant installed in an icebreaking vessel operating under diverse load conditions and ambient air temperatures requires a degree of sophistication and control complexity that the possible increase of 1% in thermal efficiency would not justify.

2.1.2.7 Supplementary Firing

Additional steam generating capacity might be obtained by utilizing the large quantities of excess air (approximately 400% over the combustion requirements of the gas turbine) contained in the exhaust gas of the gas turbine to support further combustion. Combustion under these circumstances is carried out in a very efficient manner due to the preheat contained in the air. This system is often adopted in land based industrial systems, where space, weight and plant flexibility are not significantly important.

Supplementary firing of the waste heat boilers offers many advantages including:

- additional installed horsepower with marginal capital investment,

- ability to "smooth" steam generation in boiler by activating supplementary burners during periods of low gas turbine power,

- high combustion efficiency due to combustion of pre-heated air.

Supplementary firing would not require an additional source of combustion air since the gas turbine exhaust contains sufficient oxygen to promote firing. Babcock and Wilcox Ltd. advise that 17.27 lb. of air per lb. of fuel is required to support combustion with natural gas. As Table 2 demonstrates, the LM 2500 exhaust gas flow is sufficient to support a considerable percentage of additional power through supplementary firing of the waste heat boiler without need of additional combustion air.

Supplementary firing does entail a number of drawbacks:

TABLE 2

THEORETICAL POWER AVAILABLE FROM SUPPLEMENTARY FIRING

Gas Turbine Power (Megawatts)	% Combustible Air In Exhaust Gas by Weight	Exhaust Gas Mass Flow (lb/sec)	Combustible Exhaust Gas Mass Flow (lb/sec)	Theoretical Supplementary Firing Fuel Flow (lb/sec)	Theoretical Power Supplement From Supplementary Firing (Megawatts)
22.380	71.7%	148	106	6.14	29,445
20.515	73.7%	145	107	6.20	29,733
17.904	75.1%	132	99	5.73	27,479
13.428	77.1%	121	93	5.39	25,849
.1492*	90.0%	55	50	2.90	13,908

* Idle Condition

NOTES:

1. LM 2500 exhaust gas characteristics.
2. Theoretical power determined on basis of specific fuel consumption of .56 lb/HP-HR.
3. Theoretical fuel flow determined on basis of combustible air/fuel ratio of 17.27.
4. Power available through supplementary firing is theoretical only. Practical limitations such as boiler size and weight and pressure drop at the gas turbine outlet restricts the additional power available through supplementary firing to much lower levels.

- approximately two-fold increase in boiler height,

- additional capital cost (Foster Wheeler estimate that supplementary firing increases boiler capital cost 20% - 25%),

- additional 1.5" to 2" water gage pressure drop at gas turbine exhaust,

- potential difficulties in locating burners where they will not disturb air flow yet be effective and reliable,

- need for forced air fans to provide combustion air if gas turbine is not operating,

- special attention must be provided to boiler materials as a result of radiant heat of supplemental flame.

These drawbacks, the lack of marine experience and the attendent operating problems associated with a more complex plant suggest supplementary firing should not be adopted for an icebreaker.

2.1.2.8 Dry Firing of Boiler

There will be occasions with the combined cycle propulsion system when the gas turbine will be operated without the waste heat boiler functioning. This might occur when the boiler is off-service awaiting maintenance but gas turbine power is still required.

This may be accomplished by providing a by-pass on the exhaust duct of the gas turbine prior to the boiler or by designing the boiler to be dry fired.

Exhaust gas by-pass dampers have seen considerable use in shore-based installations but have little marine experience. The shore-based units demonstrate certain characteristics which on land are not severe but on a vessel do pose difficulty. This includes:

- additional space requirements,

- noise caused by air "bleeding" past damper,

- distortion of damper due to high gas temperatures.

Boiler manufacturers recommended dry firing of the boilers be adopted provided appropriate attention is paid to materials. The materials selected for the exhaust gas boiler will satisfy Classification Society regulations, and will be suitable to avoid high temperature corrosion during dry firing.

2.1.2.9 Boiler Tubing

In order to fully accept the rapid load change characteristics of the gas turbines conducting icebreaking duties, the waste heat boilers must be specially designed. Typically, the waste heat boiler for this type of duty consists of horizontally finned tubes with a steam and water drum and assisted feedwater circulation. The volume of the steam drum must be adjusted to accept the rapid swell or contraction of the boiler water due to fluctuations in steam to water ratio. This will result from variation in boiler evaporation rate during the changes in gas turbine load.

It is suggested, therefore, that the waste heat boilers for an Arctic icebreaker be designed with horizontal tubing, assisted circulation and a larger than conventional steam drum.

2.1.2.10 Number of Waste Heat Boilers

In a multiple gas turbine installation, certain economies may be achieved by allowing two or more gas turbines to exhaust into one waste heat boiler. This concept has certain advantages including:

- lower capital cost (preliminary estimates suggest capital cost of one boiler fed by two gas turbine exhausts is approximately 80% of the capital cost of two single boilers),

- less space required (at a minimum, a saving of 6 feet in width as a result of reduced maintenance area).

However, there are disadvantages associated with multiple gas turbines exhausting into a single boiler:

- possibility of gas turbulents at exhaust duct juncture which increases gas turbine back pressure and decreases boiler efficiency,

- less efficient since smaller boiler designed for a single gas turbine exhaust flow may be optimized for one gas path,

- difficulty in installing exhaust ducts within the space limitations imposed on a marine vessel,

- requirement for complex system of dampers to avoid gas turbine exhaust backflow through unused gas turbines,

- less flexibility in terms of maintaining the boilers.

One boiler per gas turbine is suggested in view of the potential difficulties associated with boilers incorporating multiple gas turbine exhausts. The boiler industry agreed with this conclusion, detailing a number of technical difficulties they have encountered with boilers having multiple gas turbine exhausts. One manufacturer recommended one boiler per gas turbine but suggested certain economies might be realized by incorporating a common steam drum.

2.1.3 Control of the Combined Cycle System

The steam portion of the combined cycle power plant is a means of extracting energy that otherwise might be lost from gas turbine exhaust. In capturing this energy, a vessel operator does not wish to further complicate the engine room machinery or substantially increase crew to operate and maintain this equipment. Consequently, the control system for the steam plant must be:

- simple and automatic requiring little operator attention,

- reliable, allowing high availability of the steam turbine power,

- proven, thereby minimizing unexpected downtime and subsequent maintenance.

With these objectives in mind, a constant pressure steam system appears most appropriate for an Arctic icebreaker. This system of control might be selected since the steam generated by the waste heat boilers will not be dedicated to the main propulsion system. This steam will also be used for auxiliary purposes such as ship service power using a turbo-generating set. In this situation, the ship service turbo-generators will be the preferred steam customer with the balance of the steam passed to the main propulsion turbines. The ship service turbine requires a reasonably constant steam pressure to maintain turbine speed and load. Figure 11 depicts a simplified schematic diagram of this control system.

Should the main propulsion steam turbine be the sole recipient of steam produced by the waste heat boiler, throttling losses to the turbine may be minimized through the use of a sliding pressure boiler control system. With this control, boiler pressure is allowed to fluctuate, within prescribed limits, with gas turbine exhaust conditions. This control allows boiler pressure to rise during periods of low power demand minimizing steam dump to the condenser.

2.1.4 A Waste Heat Boiler for an Arctic Icebreaker

After consideration of the aforementioned factors, Table 3 depicts a waste heat boiler design and steam characteristics suitable for installation aboard an Arctic icebreaker. Figures 12 and 13 also present this information. This waste heat boiler design may be described in the

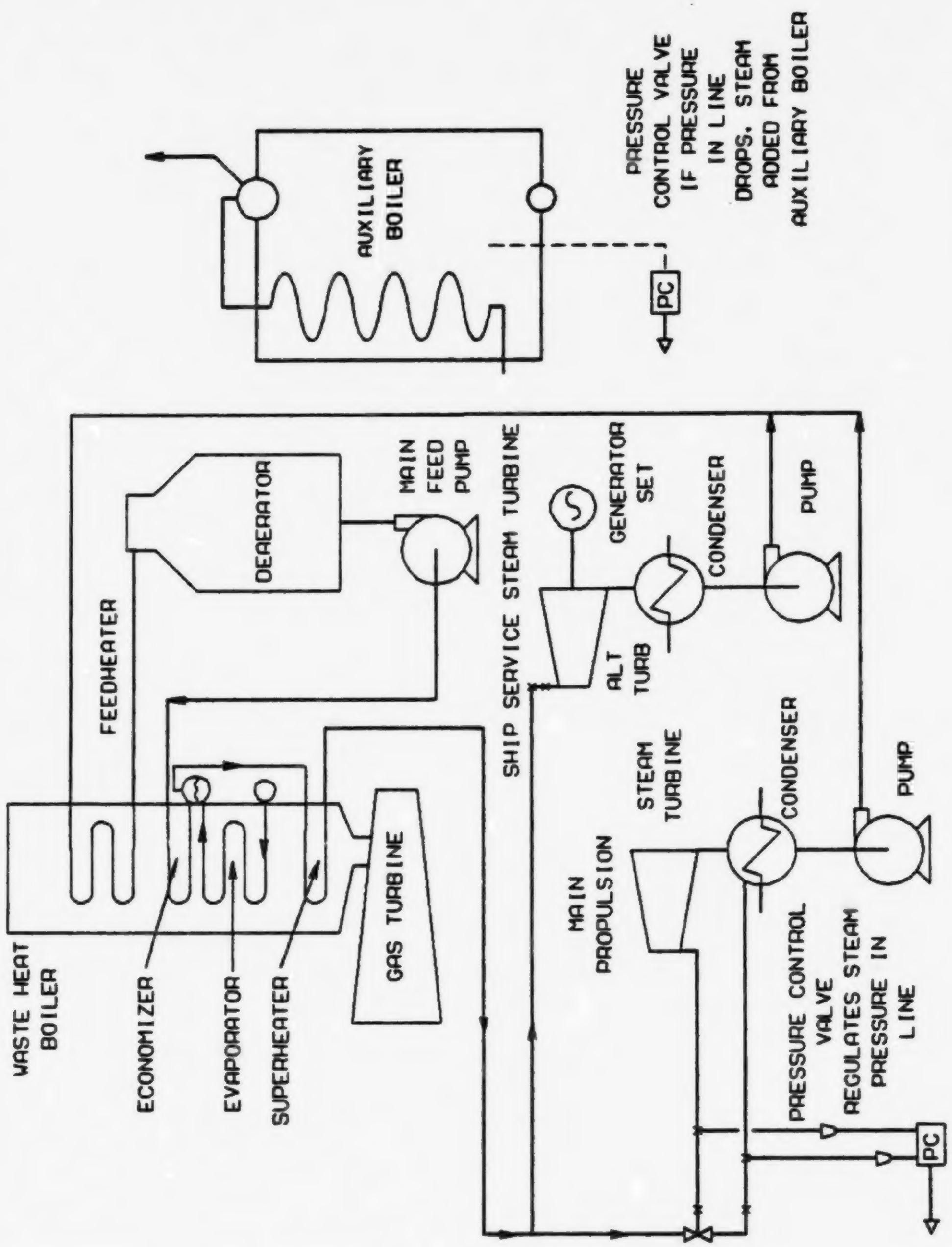

WASTE HEAT BOILER
ECONOMIZER
EVAPORATOR
SUPERHEATER
GAS TURBINE
FEEDHEATER
DEAERATOR
MAIN FEED PUMP
SHIP SERVICE STEAM TURBINE
GENERATOR SET
ALT TURB
CONDENSER
PUMP
MAIN PROPULSION
STEAM TURBINE
CONDENSER
PUMP
PRESSURE CONTROL VALVE REGULATES STEAM PRESSURE IN LINE
PC
AUXILIARY BOILER
PC
PRESSURE CONTROL VALVE IF PRESSURE IN LINE DROPS, STEAM ADDED FROM AUXILIARY BOILER

FIGURE 11
POSSIBLE CONTROL SYSTEM
FOR COGAS PROPULSION

TABLE 3
WASTE HEAT BOILER DESIGN CALCULATIONS

Ambient Air Temperature	-6.7°C (20°F)
G. E. LM 2500 Exhaust Temperature (from Figure 4)	454°C (850°F)

A. Superheater - Evaporator Design

Approach Temperature	44°C (80°F)
THEREFORE, Temperature of Steam at Superheater Outlet =	410°C (850°F - 80°F = 770°F)
Steam Pressure	2.53 MN/m^2 (25 Atmospheres)
Enthalpy of Superheated Steam at 25 Atm., abs., 410°C (from steam tables)	1403 Btu/lb.
Enthalpy of Water at 25 Atm., 225°C (saturation temp. at this pressure)	410 Btu/lb.
THEREFORE, Heat Added in Boiler =	993 Btu/lb.
"Pinch Point" at Evaporator Inlet	22°C (40°F)
THEREFORE, Exhaust Gas Temperature at Exit from Evaporator =	247°C (437°F + 40°F = 477°F)
Exhaust Gas Flow (from Figure 4)	155 lb./sec.

The Constant Pressure Specific Heat Coefficient of Air of 0.248. This represents the Change in Enthalpy of Air for Every 1°F in Temperature.

Boiler Heat Transfer Effectiveness	0.98

THEREFORE, Steam Flow = $\frac{.248 \times 155 \times (850-477)}{794} \times \underline{.98}$

= 17.7 lb./sec.

= 63,709 lb./hr.

NOTE: Steam flow proportional to exhaust gas flow.

B. Economizer Design

Economizer Water Inlet Temperature	115°C (240°F)
Enthalpy of Water at Economizer Outlet (25 Atm., 225°C)	410 Btu/lb.
Enthalpy of Water at Economizer Inlet (115°C)	209 Btu/lb.
THEREFORE, Heat Added in Economizer	201 Btu/lb.
THEREFORE, Temperature Drop of Exhaust Gases in Economizer =	$\frac{63{,}709 \times 201}{3600 \times .248 \times 155 \times .98}$
=	34.4°C (94.4°F)
THEREFORE, Temperature of Exhaust Gases at Exist from Economizer =	194°C (477°F - 94.4°F = 382.6°F)

C. Steam Turbine Design

Condenser Pressure and Temperature	1.5 lb./in^2, 43°C (110°F)
Enthalpy of Steam at Turbine Inlet (25 Atm., 427°C)	14.3 Btu/lb.
Enthalpy at Turbine Exit (1.5 lb./in^2, 43°C)	961 Btu/lb.
Theoretical Heat Drop	442 Btu/lb.
Typical Steam Turbine Efficiency	75%

THEREFORE, Actual Heat Drop = 442 x .75 = 331.5 Btu/lb.

THEREFORE, Enthalpy at Turbine Exit = 1071.5 Btu/lb.

D. Steam Turbine Output

Actual Heat Drop Through Turbine	331.5 Btu/lb.
Steam Flow	63,709 lb./hr.
Total Actual Heat Drop Through Turbine	21.12×10^6 Btu./hr.
Steam Turbine Power	8302 HP = 6193 KW

NOTES: Excludes piping and throttle losses. Steam turbine output is proportional to steam flow; therefore, steam turbine output is also proportional to gas turbine exhaust flow.

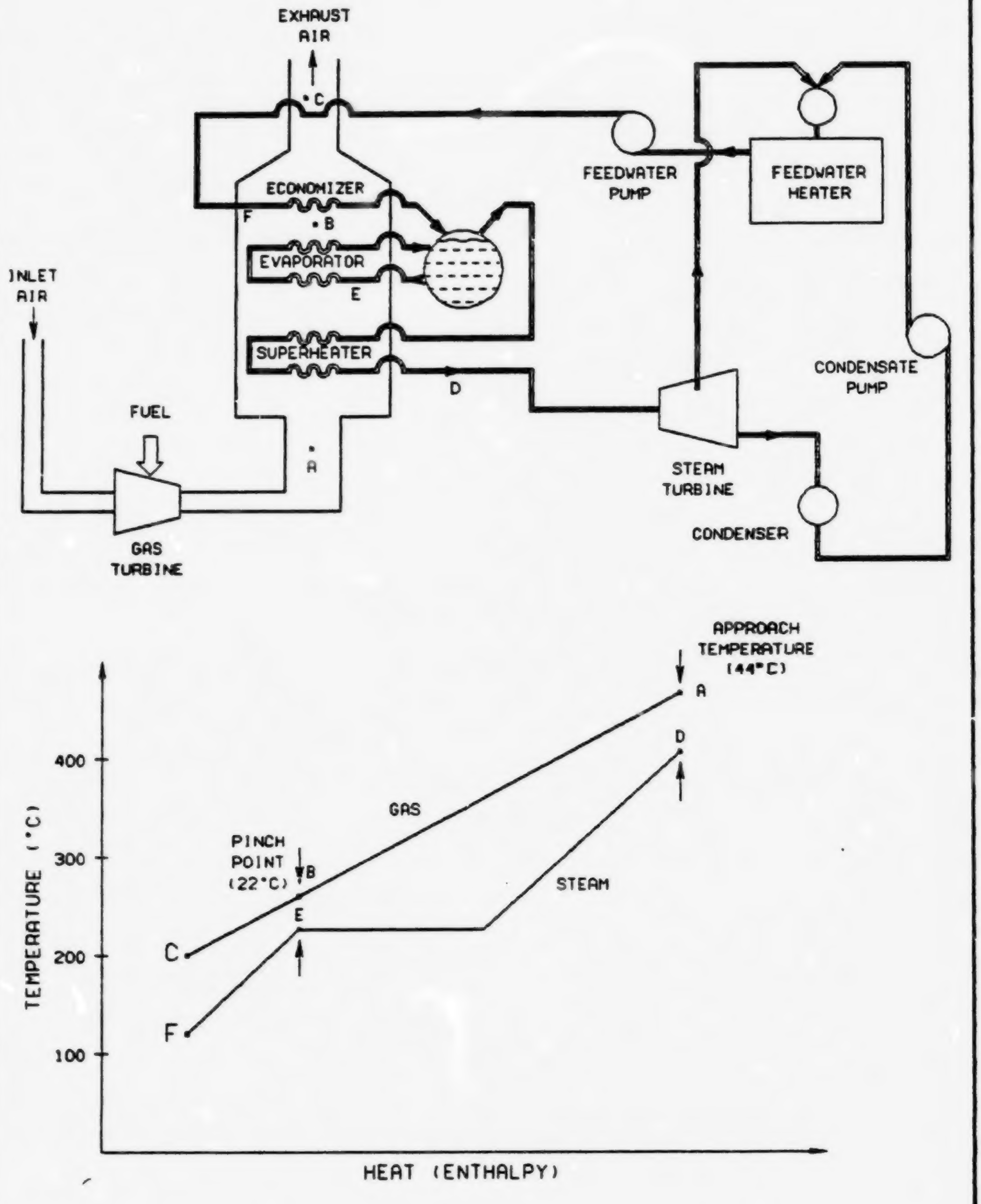

EXHAUST AIR
C
FEEDWATER PUMP
FEEDWATER HEATER
ECONOMIZER
F
B
EVAPORATOR
E
INLET AIR
SUPERHEATER
D
CONDENSATE PUMP
FUEL
A
STEAM TURBINE
GAS TURBINE
CONDENSER
APPROACH TEMPERATURE (44°C)
A
D
400
GAS
PINCH POINT (22°C)
B
300
STEAM
E
TEMPERATURE (°C)
C
200
F
100
HEAT (ENTHALPY)

FIGURE 12
TEMPERATURE-ENTHALPY DIAGRAM OF COMBINED GAS TURBINE AND STEAM TURBINE SYSTEM

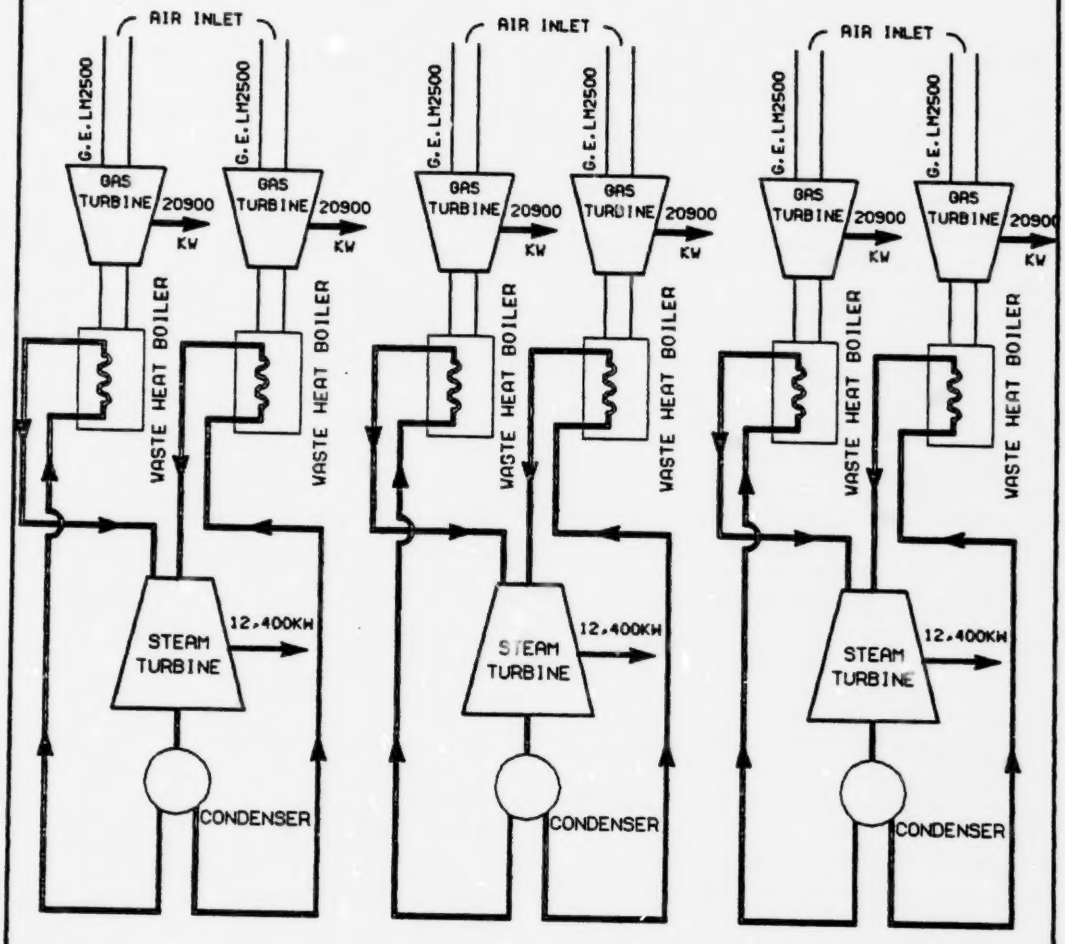

AIR INLET
AIR INLET
AIR INLET
G.E.LM2500
G.E.LM2500
G.E.LM2500
G.E.LM2500
G.E.LM2500
G.E.LM2500
GAS TURBINE
GAS TURBINE
GAS TURBINE
GAS TURBINE
GAS TURBINE
GAS TURBINE
20900 KW
20900 KW
20900 KW
20900 KW
20900 KW
20900 KW
WASTE HEAT BOILER
WASTE HEAT BOILER
WASTE HEAT BOILER
WASTE HEAT BOILER
WASTE HEAT BOILER
WASTE HEAT BOILER
STEAM TURBINE
STEAM TURBINE
STEAM TURBINE
12,400KW
12,400KW
12,400KW
CONDENSER
CONDENSER
CONDENSER

FIGURE 13
SCHEMATIC DIAGRAM OF
COMBINED AERO-DERIVED GAS TURBINE
& STEAM TURBINE PROPULSION SYSTEM

following manner:

- approach temperature 44°C (80°F),
- pinch point 22°C (40°F),
- exhaust stack temperature 127°C (260°F) minimum,
- single pressure,
- no supplementary firing,
- capable of dry firing,
- horizontal tubing,
- assisted circulation,
- designed for operation with one gas turbine only.

2.1.5 Steam Turbine, Condenser and Deaerator

The steam turbine will be designed for the steam conditions selected. For a COGAS system designed to operate primarily in open water, the minimum ambient air conditions which the plant may be expected to experience would probably be in the region of 0°C. In this case, a steam temperature at the superheater outlet of 500°C might be selected to attain maximum efficiency of the steam cycle.

In the case of a vessel operating in an Arctic environment, with ambient air temperatures as low as -55°C, the maximum steam temperature attainable would only be approximately 350°C.

As a result, the design of the steam turbine would be orientated, not so much towards the maximum attainable efficiency, but to simplicity of design and maximum availability. The turbine selected would be designed to operate satisfactorily with low pressure, low temperature steam, due to the constraints imposed by the low ambient air conditions and the resultant low exhaust temperatures of the gas turbine. The turbine casing and blading might be constructed from carbon or low alloy steels. The internal clearances of the turbine would be made larger than is normal to prevent interference between the moving and stationary parts which might occur due to the possibility of ingress of wet steam when operating in Arctic conditions. Since it is impractical and inefficient to design the steam system for the lowest possible Arctic ambient temperature, the COGAS system will occasionally be required to operate in ambient temperatures below the design point. Under these conditions, the last states of the steam turbine may become extremely "wet", which may require water extraction stages to be incorporated in the turbine design together with stellite shielding of the last stage blading if blade and casing erosion is to be avoided.

The turbine might be arranged with a conventional under-slung condenser or as an axial flow unit. The axial flow units have become increasingly popular in recent times in part due to the advent of epicyclic gearing in the marine industry and in part to the 2% - 3% improvement in turbine efficiency resulting from a reduction in the leaving losses. The disadvantages of these condensers are the extra length of engine room required and the relative inaccessibility of the forward turbine bearing.

The condenser will be conventional construction of the "tube in shell" type, having cast iron or mild steel fabricated water boxes, suitably protected against corrosion, with rolled naval brass tube plates fitted with 70/30 cupronickel tubes. The condenser would be arranged to operate under vacuum conditions with the ability to vary the vacuum dependent upon the steam conditions admitted to the turbine. When low temperature steam is being generated in low ambient temperatures, the condenser pressure should be raised to increase the turbine exhaust temperature and thus reduce the wetness fraction of the steam. It might be advantageous to use in this application a mechanical air pump to evacuate the condenser rather than a steam jet type air ejector.

In the case of a waste heat steam system, it is uneconomical to bleed steam from the steam turbine for feed heating purposes. The condensate from the condenser would therefore be transferred from the condenser directly to a deaerator (direct contact feed heater). The steam for this feed heater might be supplied from the waste heat boiler or preferably from a feed water heat exchanger fitted after the boiler economizer to provide deaeration duties. It should be noted that although the deaerator acts as a feed heater, its primary duty is to remove oxygen from the boiler feed water and thus prevent boiler and steam system corrosion.

3. TECHNICAL EVALUATION OF THE COMBINED CYCLE SYSTEM

One method of determining the technical suitability of a piece of equipment or system to a particular application is to simulate the equipment undertaking the proposed operating duty. The Arctic Pilot Project has conducted dynamic analyses using hybrid analogue and digital computers of many propulsion systems conducting icebreaking manoeuvers. The hybrid computers simulate system control in minute detail while monitoring various operating parameters to gauge vessel performance. The evaluation and interpretation of these results then provides certain indication as to the effectiveness of the system for the considered application.

The evaluation of an icebreaker propulsion system might include simulation of duties such as:

- crash stop,
- ice ramming,
- ice milling by the propeller,
- and acceleration.

Unfortunately, detailed dynamic simulation of a propulsion system is very costly and outside the financial bounds of this study.

Solar Turbines International of San Diego, California have developed a combined cycle model capable of simulating the General Electric LM 2500 gas turbine and a waste heat boiler of their own design. Solar agreed to simulate, with this model, a combined cycle propulsion system conducting an ice ramming cycle, perhaps the most arduous of icebreaking operations.

Solar have digitized the exhaust gas characteristics of the LM 2500 gas turbine. This allows the model to accept gas turbine load as input and then internally determining exhaust gas temperature and mass flow at the entrance to the boiler.

The Solar waste heat boiler is a "once through", dual pressure, finned tubed boiler shown schematically in Figure 14. The water/steam cycle is essentially closed loop with minimal make-up feed-water required.

The waste heat boiler control is somewhat unconventional. The following paragraphs, excerpted from a paper presented to the San Diego chapter of the Society of Naval Architects and Marine Engineers in January, 1979, describes Solar's proposed control system:

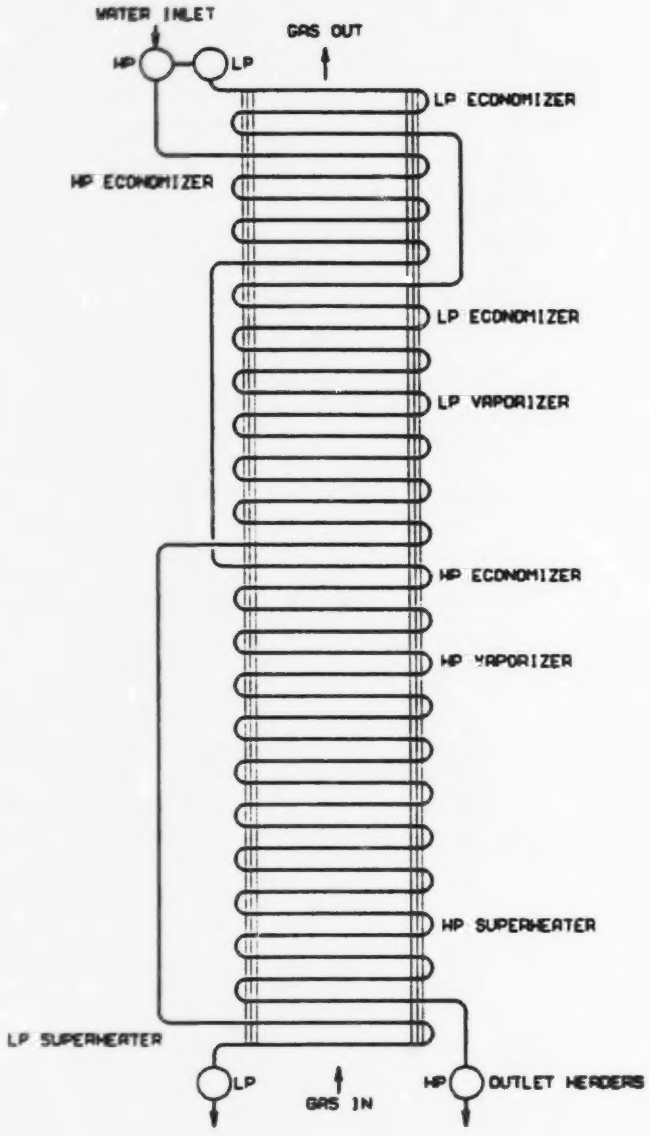

WATER INLET
HP
LP
GAS OUT
LP ECONOMIZER
HP ECONOMIZER
LP ECONOMIZER
LP VAPORIZER
HP ECONOMIZER
HP VAPORIZER
HP SUPERHEATER
LP SUPERHEATER
LP
GAS IN
HP
OUTLET HEADERS

FIGURE 14
SCHEMATIC DIAGRAM OF
SOLAR TURBINES INTERNATIONAL
"ONCE-THROUGH" WASTE HEAT BOILER

> "The steam cycle is designed to operate as a slave to the gas turbine. This means that boiler pressure will be allowed to vary as gas turbine power varies. This approach eliminates the need for boiler pressure control and permits the use of a full admission steam turbine, thereby also eliminating the losses normally experienced with typical segmented nozzle partial arc admission turbines.
>
> The once-through boiler has comparatively simple control requirements which lead to a simple overall control system. A solid state electronic control module using microprocessor technology has been developed. Two means of boiler control are possible; predictive control of superheater temperature and closed loop control based on boiler gas-side approach temperature.
>
> In the first case, the thermodynamic analysis of the boiler operation indicates that effective control of the feedwater inlet flow rate may be based on enthalpy balance. This means that, with a fixed outlet restriction and a specific input gas enthalpy, the output steam pressure and temperature is directly related to feedwater flow rate. Th microprocessor controller uses the predictive energy balance equation to establish an output parameter for use in controlling feedwater control valve position.
>
> In the closed loop operating mode, control is based on the difference between the high pressure steam temperature leaving the boiler and the temperature of the gas turbine exhaust entering the gas side of the boiler. Feedwater flow is regulated to maintain the desired temperature difference."

Figure 15 depicts the microprocessor controller operation.

As mentioned, Solar agreed to simulate an icebreaking manoeuver using their computer model of the COGAS system. The duty selected for simulation was the ice-ramming cycle which is perhaps the most arduous manoeuver for the propulsion equipment of an icebreaker. In conducting the ice-ramming manoeuver, the propulsion equipment completes two thermal cycles. The waste heat boiler also experiences significant fluctuation in the gas turbine exhaust air mass flow and temperature available to the boiler.

Figure 16 outlines the ice-ramming cycle provided to Solar by the Arctic Pilot Project. This ramming cycle consists of:

- vessel is proceeding at full power in ice-covered waters,

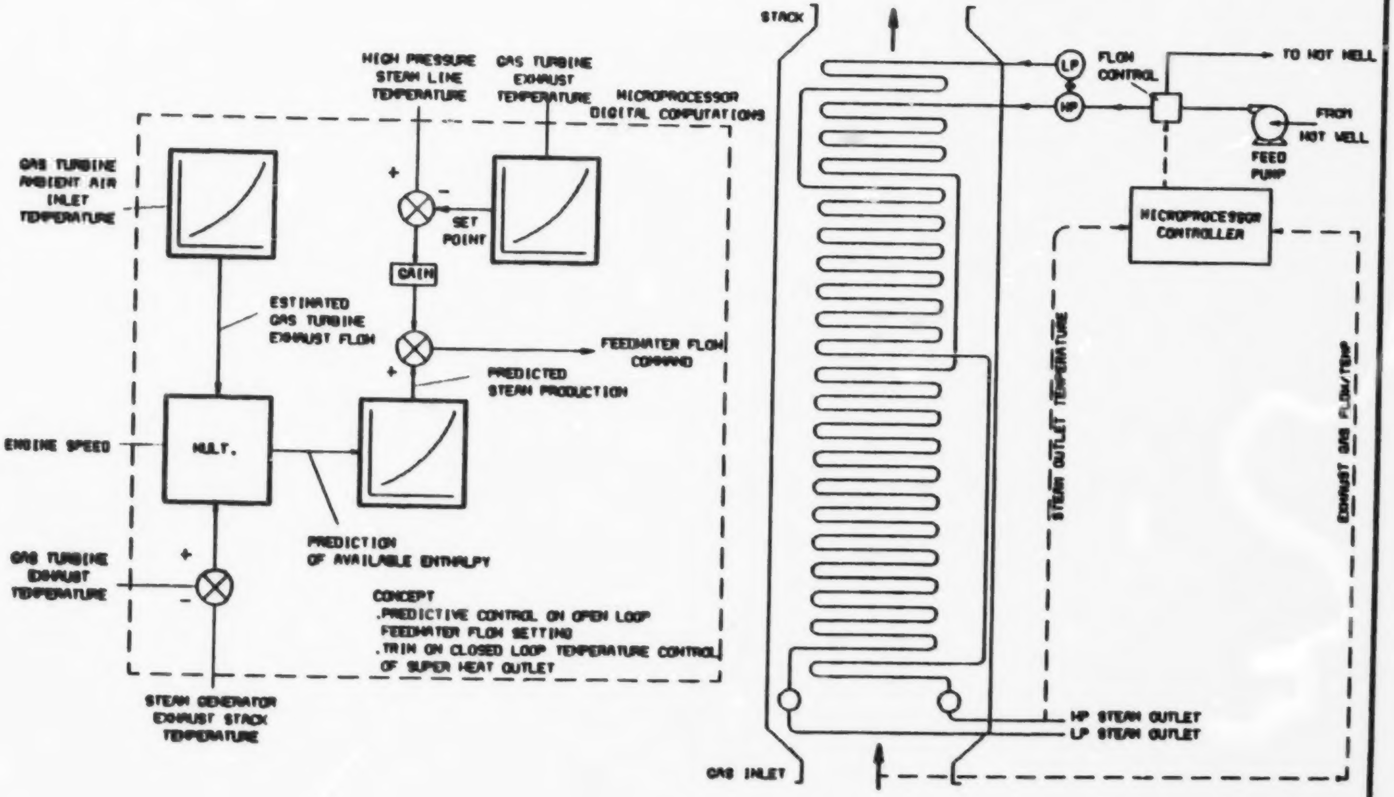
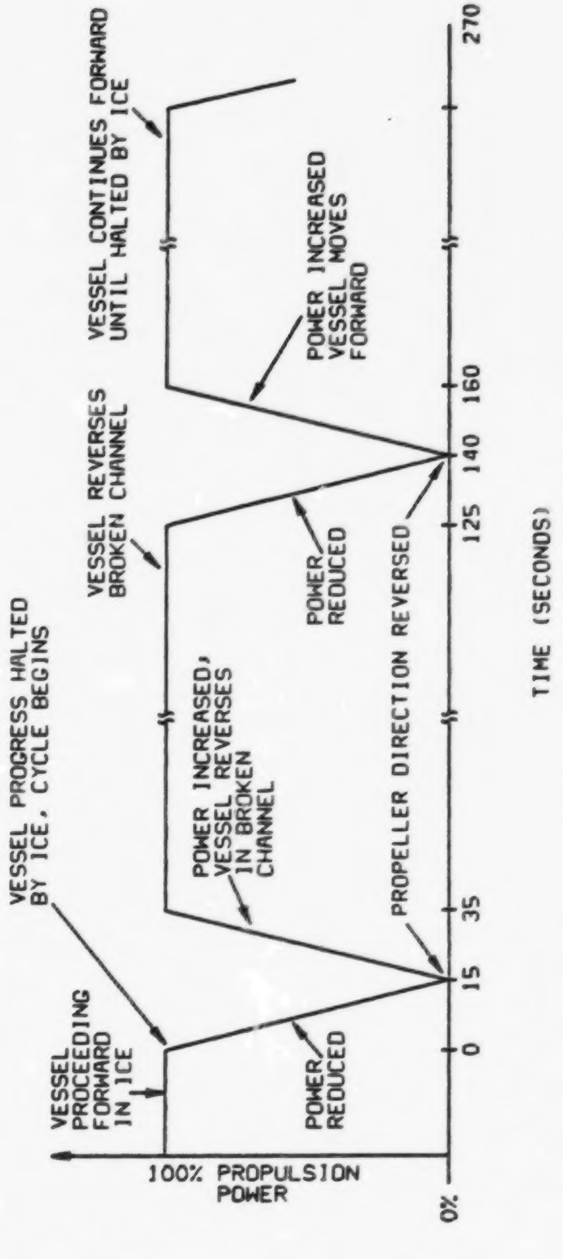

STACK
GAS INLET
LP
HP
FLOW CONTROL
TO HOT WELL
FROM HOT WELL
FEED PUMP
MICROPROCESSOR CONTROLLER
STEAM OUTLET TEMPERATURE
EXHAUST GAS FLOW/TEMP
HP STEAM OUTLET
LP STEAM OUTLET
HIGH PRESSURE STEAM LINE TEMPERATURE
GAS TURBINE EXHAUST TEMPERATURE
MICROPROCESSOR DIGITAL COMPUTATIONS
GAS TURBINE AMBIENT AIR INLET TEMPERATURE
+
-
SET POINT
GAIN
+
ESTIMATED GAS TURBINE EXHAUST FLOW
FEEDWATER FLOW COMMAND
PREDICTED STEAM PRODUCTION
ENGINE SPEED
MULT.
PREDICTION OF AVAILABLE ENTHALPY
GAS TURBINE EXHAUST TEMPERATURE
+
-
CONCEPT
.PREDICTIVE CONTROL ON OPEN LOOP FEEDWATER FLOW SETTING
.TRIM ON CLOSED LOOP TEMPERATURE CONTROL OF SUPER HEAT OUTLET
STEAM GENERATOR EXHAUST STACK TEMPERATURE

FIGURE 15
CONTROL SYSTEM FOR
SOLAR TURBINES
INTERNATIONAL WASTE HEAT BOILER



NOTE: IN PRACTICE, FULL POWER ASTERN IS NOT REQUIRED

FIGURE 16
SIMULATED ICE RAMMING CYCLE

- vessel progress is halted by an ice feature; cycle commences,

- power is reduced from full to 5% throttle in a period of 15 seconds; 5% power is absorbed in maintaining the inertia of the prime mover and generator sets and in the braking resistors,

- instantaneous reversal of motor field (in practice, this procedure requires a few seconds),

- increase in power from 5% to full throttle in 20 seconds; vessel begins to reverse in broken ice channel,

- full power for a period of 90 seconds; vessel continues to reverse in the broken ice channel,

- reduction in power from full to 5% throttle in a period of 15 seconds, vessel has completed its reversal in the broken ice channel and now prepares to proceed forward,

- instantaneous reversal of motor field,

-increase in power from 5% to full throttle in a period of 20 seconds; vessel prepares to move forward in broken ice channel,

- full power as vessel proceeds ahead in broken ice channel until again contacting ice feature; ramming cycle complete.

The ramming cycle is considered onerous for an icebreaking vessel since the prime movers, in a limited period of time, complete four excursions from a low power to full power situation. For a COGAS propulsion system, this raises concerns such as:

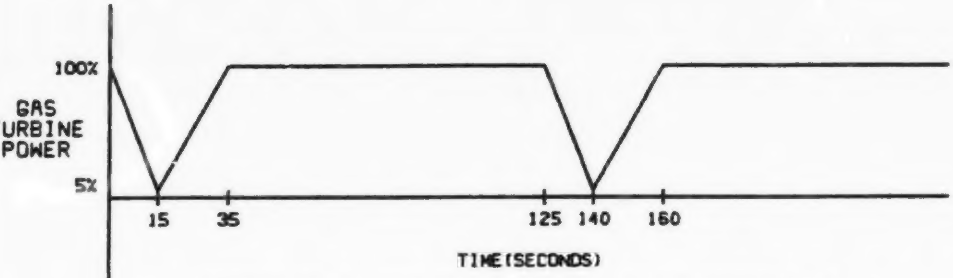
- thermal cycle fatigue in gas turbine, waste heat boiler or steam turbine,

- ability of boiler to generate steam rapidly as gas turbine power increases,

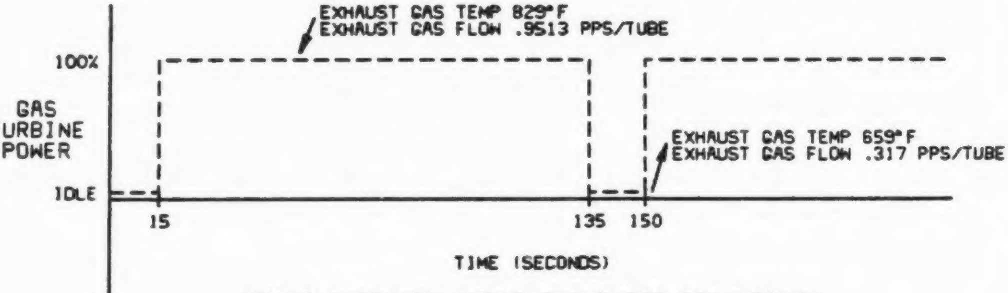
- internal operation of the boiler during gas turbine power fluctuation,

- ability of the propulsion system to complete the ramming cycle in the time specified due to lagging steam power.

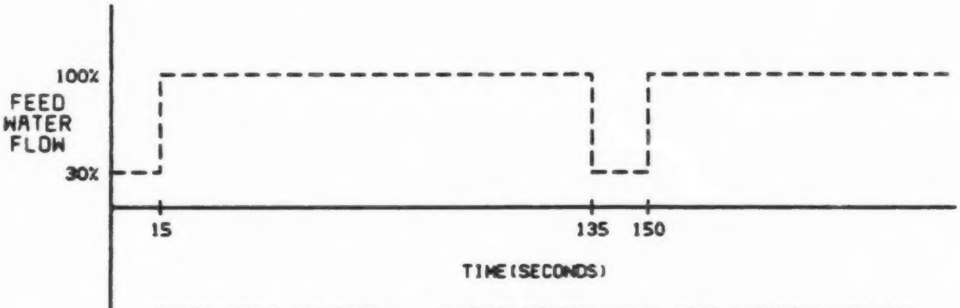
Unfortunately, the Solar model was not sufficiently sophisticated to accept a ramp function input representing the gas turbine power. The input was therefore a step function, as shown in Figure 17, which creates a more arduous environment for the waste heat boiler. The simulation



ARCTIC PILOT PROJECT PROPOSED ICE RAMMING CYCLE



SOLAR TURBINES INPUT CYCLE FOR SIMULATION



FEEDWATER CONTROL - INTERLOCKED WITH GAS TURBINE POWER

FIGURE 17
INPUT TO THE SOLAR TURBINES
INTERNATIONAL SIMULATION
OF ICE RAMMING CYCLE

outputs monitored were solely waste heat boiler characteristics and consisted of:

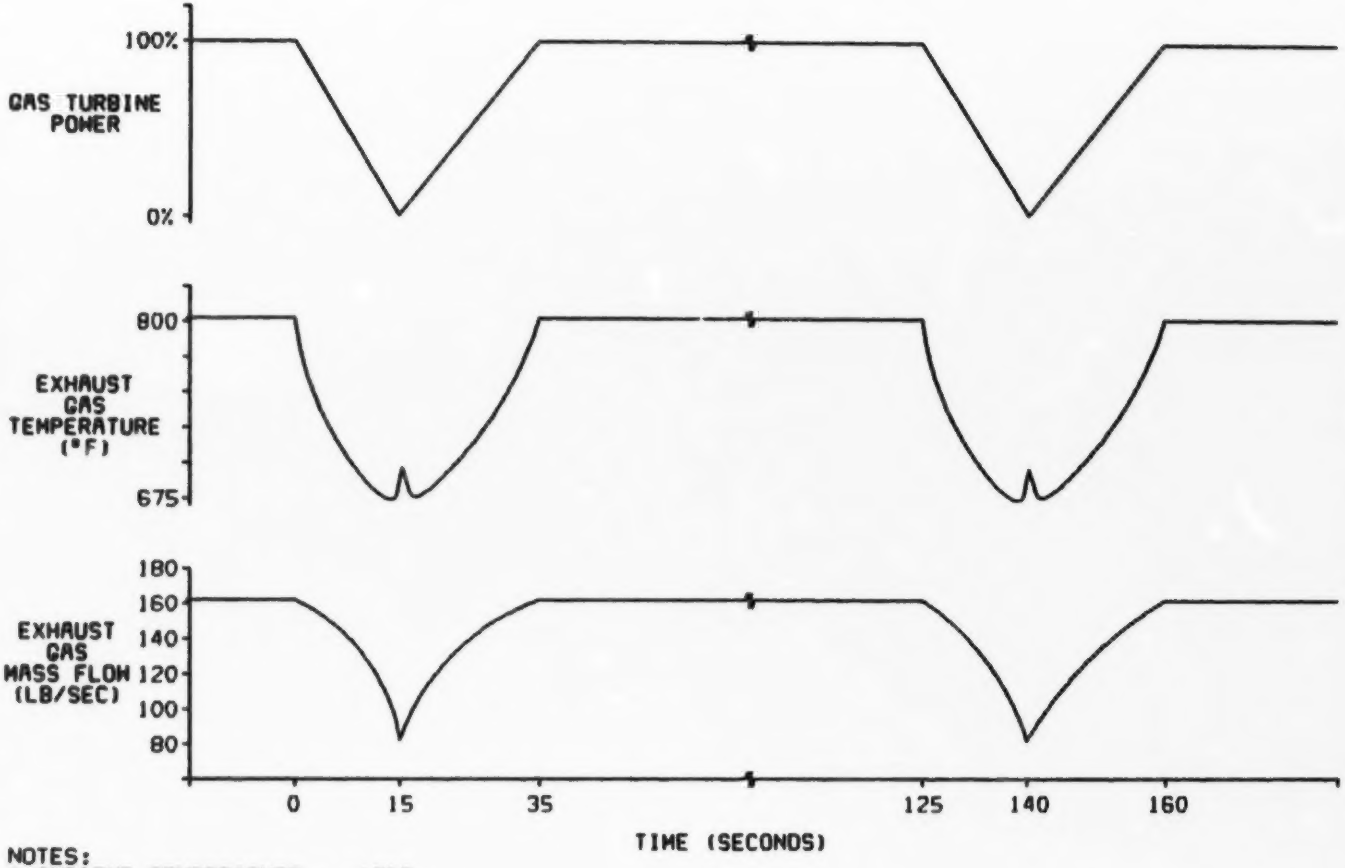
- feedwater flow,
- location of dryout zone (position where steam first becomes saturated),
- high pressure steam temperature,
- high pressure steam flow,
- low pressure steam temperature,
- and low pressure steam flow.

Figure 18 describes the variation in LM 2500 exhaust gas temperature and mass flow during the ramming cycle. This limited fluctuation in temperature will not cause difficulty to the waste heat boiler as the current "state of the art" thin wall tubing design will absorb the minor expansions/contractions occurring during this period. Easily, the greatest strain will occur during cold start-up of the boiler but this will not present severe stress provided reasonable precautions are undertaken.

Figure 19 describes the feedwater flow to the boiler and the steam flow at the outlet of the superheater. As mentioned, the feedwater flow in the Solar "once-through" boiler is predicated upon the power level and predicted exhaust gas characteristics of the gas turbine. With an immediate drop in gas turbine power, there is a corresponding decline in feedwater flow. The flow is maintained at this level until gas turbine power is again increased.

As feedwater flow is reduced with the initial decrease in gas turbine power, both high pressure and low pressure steam flow from the boiler surges as the pressure relief valve at the outlet strives to maintain pressure drop across the boiler. The steam flow eventually declines below the full power production. With increase in gas turbine power to full load, there is a slight lag in steam flow before increasing and then reaching equilibrium approximately one minute later. Figure 20 describes the variation in low pressure and high pressure steam temperature during the ramming cycle.

Figure 21 depicts the movement of the "dry out"zone during this period. Should the dryout zone move drastically within the boiler, this portends of potential control difficulties. In particular, there is the possibility of a "wet" steam entering the turbine with subsequent erosion of blades.



NOTES:
1.AMBIENT TEMPERATURE : -15°F
2.DUCT LOSSES ARE NOT CONSIDERED

FIGURE 18
DESCRIPTION OF LM 2500
ESTIMATED EXHAUST GAS
CHARACTERISTICS DURING
ICE RAMMING CYCLE

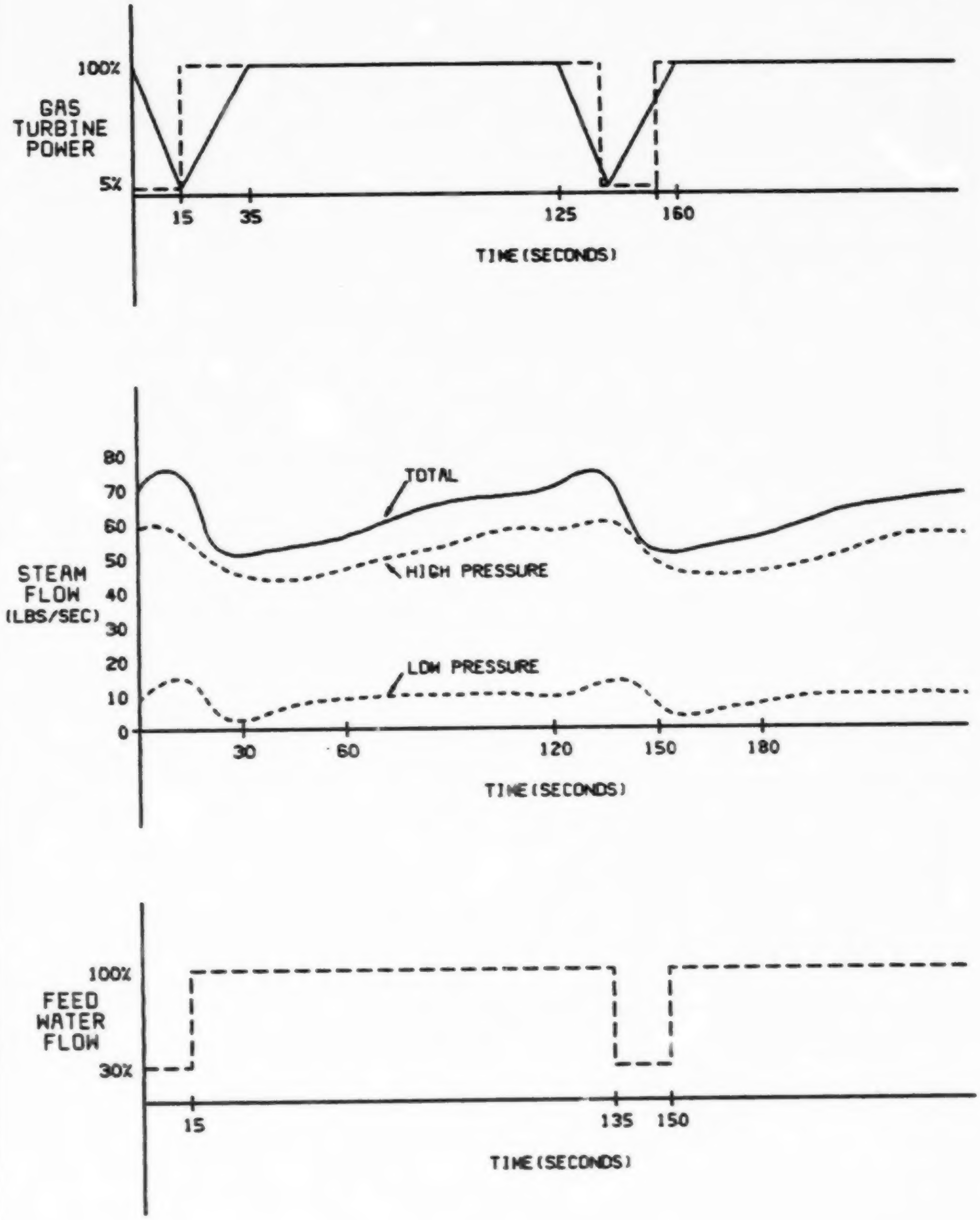


FIGURE 19
RESULTS OF SOLAR TURBINES INTERNATIONAL SIMULATION OF ICE RAMMING CYCLE

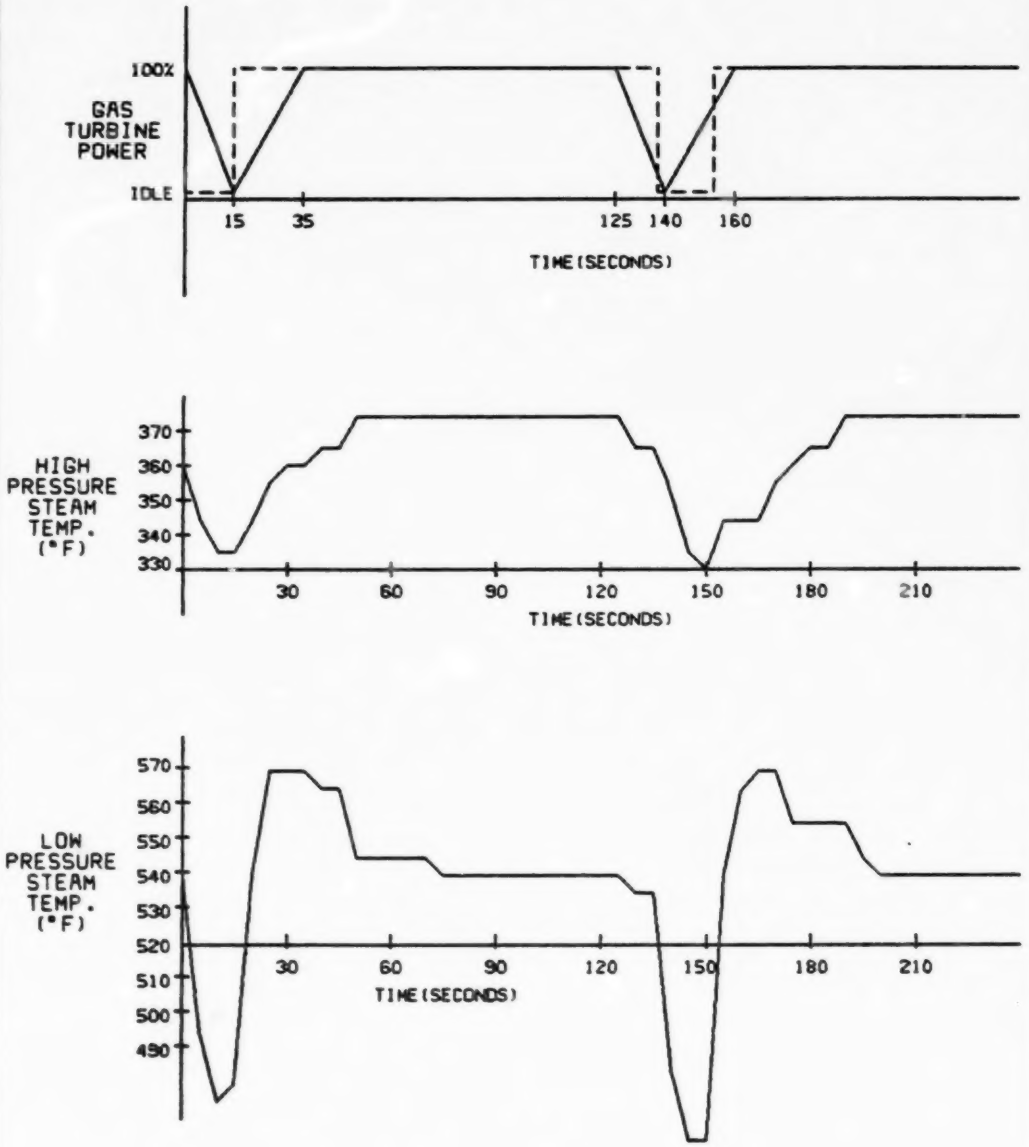


FIGURE 20
RESULTS OF SOLAR TURBINES INTERNATIONAL SIMULATION OF ICE RAMMING CYCLE

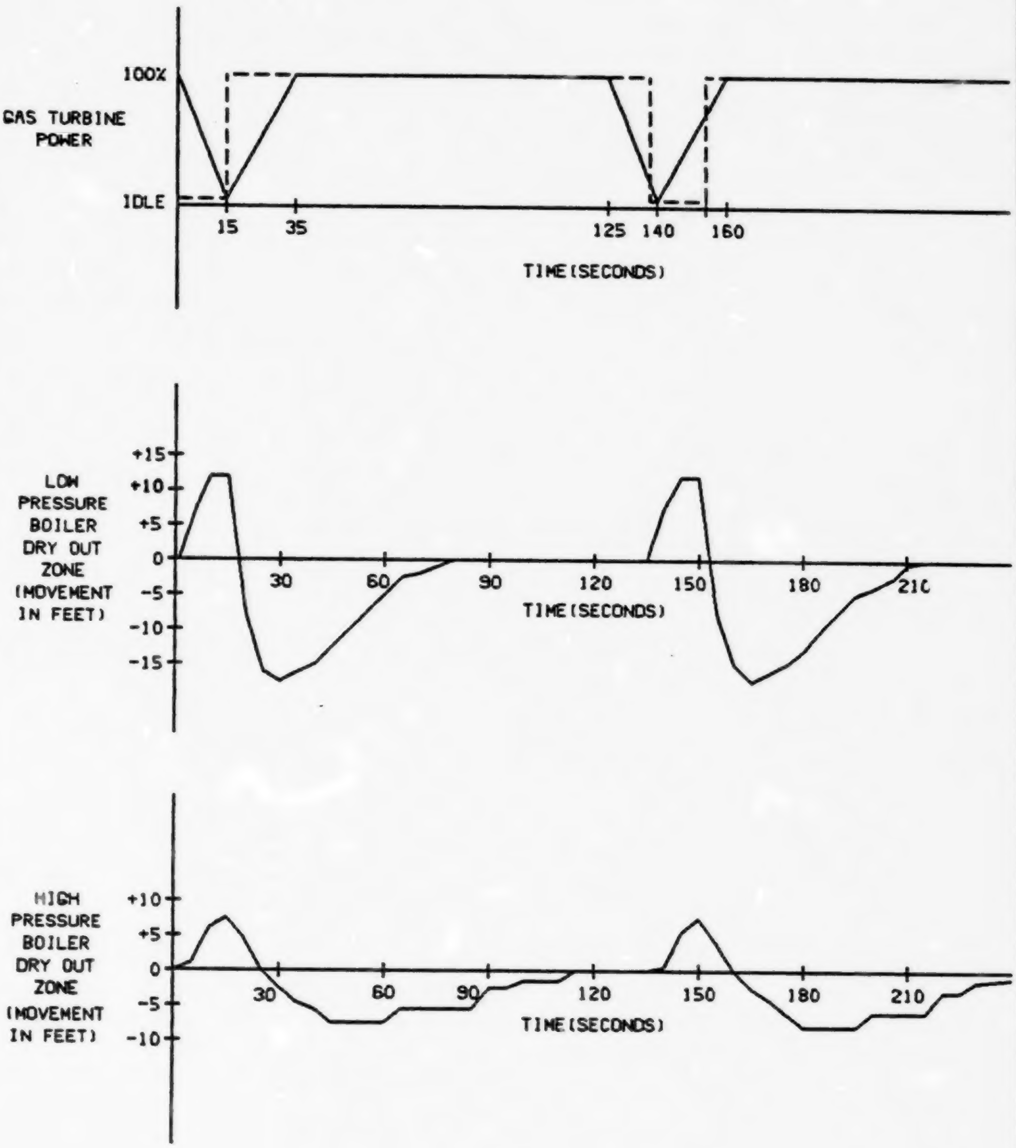


FIGURE 21
RESULTS OF SOLAR TURBINES
INTERNATIONAL SIMULATION
OF ICE RAMMING CYCLE

The simulation results suggest the dryout zone will not travel seriously meaning the "once-through" boiler is stable during the ice-ramming cycle.

The Solar simulation results indicate that the high pressure and low pressure steam systems do not experience significant fluctuation during the ice-ramming cycle. Also, as Figure 21 describes, the COGAS system has returned to equilibrium prior to contacting the ice edge. Full power is available when needed most, as the vessel contacts the ice feature which necessitated ramming in the first instance.

It is difficult to apply these simulation results to a combined cycle system having a conventional single-pressure boiler with steam drum. However, the presence of the steam drum will tend to dampen the steam fluctuations during the ramming cycle.

Figure 22 describes the estimated fluctuations in power of a conventional COGAS plant during an ice-ramming cycle. This figure was developed primarily from data presented in a paper entitled "A Dynamic Analysis of a COGAS Propulsion Plant". This paper presented the results of a dynamic simulation of a COGAS system comprised of a General Electric LM 2500 and a conventional single pressure waste heat boiler with a steam drum. The dynamic analysis concluded that:

- steam turbine power does not fluctuate rapidly with changes in gas turbine power,

- rapid excursions in gas turbine power are not "seen" by the steam system,

- the COGAS system is a "stable system capable of power excursions of over 15 MW (gas turbine) in seconds with an additional 5 MW (steam turbine) which takes 2 to 5 minutes to respond.

Figure 22 depicts an indication of COGAS power fluctuation during this ice ramming cycle, utilizing the data contained in the aforementioned paper. As gas turbine power decreases, available steam turbine power slowly decreases, available steam turbine power slowly decreases. Gas turbine power eventually increases returning to full power during the cycle but the steam turbine response is delayed as a result of its inherent thermal lag. The steam turbine eventually returns to full power well before contact with the ice edge.

It must be emphasized that Figure 22 is an approximation only of COGAS power during the ice ramming cycle. A more detailed evaluation is necessary to further gauge with confidence COGAS response during this manoeuver.

The steam turbine also has potential for experiencing certain thermal cycling during the ice-ramming cycle.

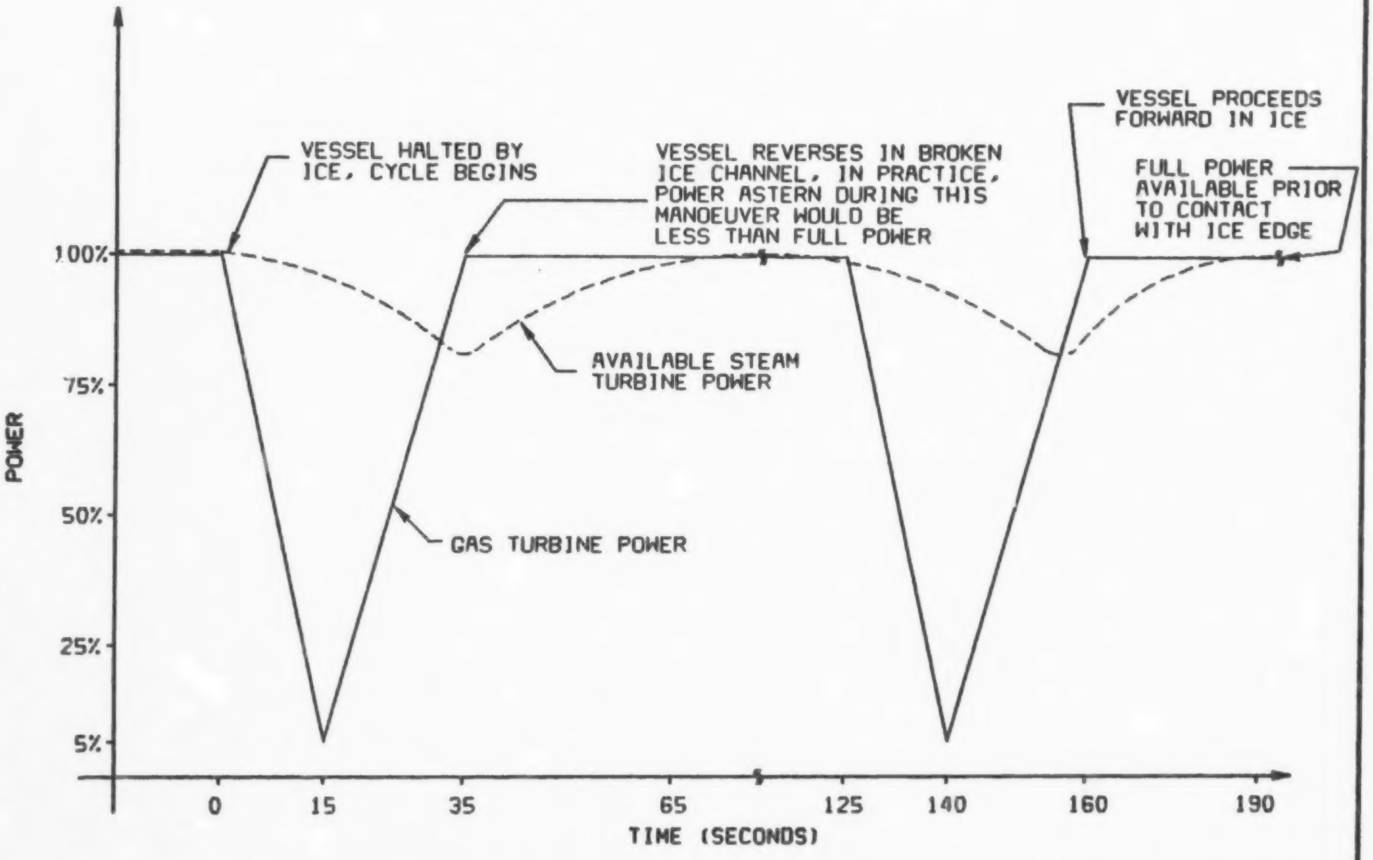


FIGURE 22
ESTIMATED FLUCTUATION OF COGAS POWER DURING THE ICE RAMMING CYCLE

G. E. C. Turbine Generators Limited evaluated for this study the effects of the ramming cycle upon the steam turbine and concluded that the cycles will not cause problems largely since the cycle times are so short. G. E. C. suggests the most critical component of the steam turbine, in terms of thermal stress, is the steam turbine rotor as the position of the high pressure gland. At this location, metal temperatures are highest in the turbine. G. E. C. studies have shown turbine stress to be a function of rotor diameter, material and mode of operation. G. E. C.'s evaluation of the proposed icebreaking ramming cycle and the resulting steam flows concluded that "the critical rotor surface temperatures will be virtually unaffected and hence thermal stress will be very low".

These studies, albeit brief, do depict a tendency concerning the ability of combined cycle to withstand the rigors of Arctic icebreaking. The outstanding uncertainties concerning this propulsion system include:

- the ability of the system to withstand the ice ramming cycle,

- the ability of the system to deliver maximum power when required,

- the ability of the system to perform reliably in an icebreaking application.

The Solar simulation also indicates that the combined cycle system will deliver maximum power when most required; upon contact with the ice edge during ramming. Although power availability does lag more conventional diesel, steam turbine or gas turbine systems, the simulation does demonstrate this lag will not hamper operations. Appropriate combined cycle control design for Arctic operation should reduce the lag in power response.

The third issue is somewhat more difficult to address. Each of the components of the combined cycle system have operated safely and reliably for many years in land-based and marine plants. By extrapolation, it may be implied that a combined cycle system installed on an icebreaker should also operate reliably but this issue merits much more detailed consideration. The following chapter addresses this question of combined cycle reliability.

4. THE RELIABILITY OF A COMBINED CYCLE SYSTEM

With the creation of the oil producing cartel in 1960 and the subsequent dramatic oil price increases in the 1970's, the installation of waste heat boilers to improve the efficiency of a fuel burning system became prominent. The fuel cost savings available with these high oil prices justified the additional capital and operating costs of a waste heat boiler. Today, there are many land-based gas turbine systems which utilize waste heat boilers to improve efficiency. As well, marine vessels typically are equipped with waste heat boilers although the steam generated, with the exception of the aforementioned Russian vessel, is used for ship service purposes.

The lack of marine experience of the COGAS propulsion system requires that assessment of reliability be subject to extrapolation of existing land-based and marine experience. This chapter will assess in this manner, the reliability of a COGAS propulsion system installed on an icebreaking vessel.

4.1 Land-Based COGAS Experience

There exists, in North America and Europe primarily, a great many land-based COGAS systems. Typically, the system is owned and operated by a utility responsible for power generation with the steam generated by the boilers being utilized for heating service or additional power generation.

The manufacturers of large aero-derived gas turbines and boilers were contacted to ascertain their understanding of the reliability of COGAS systems. The suppliers contacted included General Electric Co. Ltd. (supplier of gas turbines and boilers), United Technologies, Rolls Royce Canada Limited, Foster Wheeler Limited, and Babcock and Wilcox Limited. These suppliers were consistent in advising that the COGAS plants with which they had been associated had achieved excellent reliability. As an example, Rolls Royce cited the three combined cycle plants with which they have been associated. Rolls Royce estimated these plants had achieved greater than 95% availability. However, each of the manufacturers contacted qualified these statements by adding that typically they are informed only of major breakdowns.

A few operators were contacted in the course of this study to ascertain their experience with combined cycle systems. In each instance, the operators tended to be utilities who utilize the combined cycle plants under base load operation. Under this regime, the operators have experienced excellent availability of the combined cycle systems which might be expected. Unfortunately, detailed figures concerning the reliability of these plants were not available. In any event, there would be little relationship between these figures and those which might be anticipated for a marine plant.

4.2 Marine Waste Heat Boiler Experience

Most marinized aero-derived gas turbines are installed in vessels operated by the navies of North America and Europe. Typically, these vessels are equipped with gas turbines for cruise power, boost power and ship service power. The ship service gas turbines usually have a waste heat boiler on the exhaust stack to produce ship service hot water. These boilers bear little resemblance to the waste heat boiler of an icebreaker's COGAS system but nonetheless, it is believed they offer the only operating experience of a boiler installed on the exhaust stack of a marinized gas turbine.

In the course of this study, both the Canadian and U. S. Navies were consulted to determine their experience with waste heat boilers installed at the gas turbine exhaust outlet. In each instance, the waste heat boiler was a compact unit that was virtually an afterthought in the design stage. The boilers had not achieved what might be deemed successful operating experience but the engineers aboard these vessels felt that this was attributable more to faulty design than any inherent weakness in waste heat boilers. Among the problems cited were:

- boilers located in confined and/or inaccessible areas which do not allow routine maintenance inspection,

- improper design which provided poor gas flow and heat transfer,

- improper feedwater treatment and control.

In each discussion, the ship's engineer stated that although the boiler had experienced difficulties, he would have few qualms concerning installation of this equipment in a new building.

4.3 Marine Boiler and Steam Turbine Experience

It is difficult to assess the reliability of a waste heat boiler through the operating experience of an oil-fired boiler. However, it is important to note that operating experience with oil-fired boilers has been successful. Waste heat boilers should achieve similar, if not better, reliability since they are convection boilers utilizing conservative steam pressures and temperatures.

The reliability of marine boiler and steam turbine systems might be gauged through review of incidents recorded by the Classification Societies. Appendix 1 describes the incidents recorded in the files of Lloyd's Registry of

Shipping for:

- main membrane wall boilers developing a minimum of 15 MW power and 4 MN/m^2 steam pressure,

- steam turbines and condensers producing a minimum of 15 MW power,

- vessels of Lloyd's Register class constructed between January, 1960 and December, 1979.

The primary rationale for selecting this minimum power and pressure level was to restrict the survey to relatively modern machinery similar to that considered for an icebreaking vessel. The survey period was January, 1970 to December, 1979 inclusive.

The incidents recorded in Appendix 1 include many occasions which did not result in equipment "downtime". For instance, many steam turbine bearings and blade replacements itemized in this listing likely occurred during routine machinery survey and would not result in additional unscheduled maintenance of equipment outage. Lloyd's Register, as a matter of procedure, records each incident of this type which occurs during a routine inspection. These incidents are then recorded in their files to be itemized in a survey such as this. It is necessary, therefore, to make educated judgements as to which incidents may have resulted in unscheduled maintenance.

Appendix 1 also identifies the incidents which might have created unscheduled "downtime" of the propulsion machinery. This survey of incidents creating potential "downtime" is summarized in Table 4. As this table describes, the survey suggests the mean time between incidents for boilers generating steam of 4 MN/m^2 pressure and 15 MW power is approximately ten years. Steam turbines, producing 15 MW power or better, also have a mean time between incidents of approximately ten years. Condensers appear to have a mean time between incidents substantially greater than ten years. This survey, threfore, suggests a steam system, of the characteristics described, has experienced excellent operating success. It is difficult to assess the relationship between the reliability achieved by a steam plant installed in a conventional vessel, plying an open water route, and the reliability of a waste heat boiler/steam turbine system installed in a Canadian Arctic icebreaker. Among the factors suggesting a boiler/turbine plant installed on an icebreaker will not achieve as high reliability are:

- greater fluctuation in power levels and resulting greater variation in metal temperatures,

- potential for great vibration and mechanical stress resulting from icebreaking duty.

TABLE 4
SURVEY OF BOILER, STEAM TURBINE AND CONDENSER INCIDENTS

POWER (MW)	EQUIPMENT	NUMBER OF INCIDENTS	NUMBER AT RISK	MONTHS OF SERVICE	INCIDENCE RATE	REMARKS
15 - 22.5	Steam Turbine	88	236	11,216	1/10.6 yrs	Most defects occur in bearings, valves, blading
15 - 22.5	Condensers	17	236	11,216	1/55.0 yrs	Most downtime results from tube failure
15 - 22.5	Boilers	22	25	1,166	1/4.4 yrs	Tube failures constitute greatest incidents
22.5 - 30	Steam Turbines	82	338	13,053	1/13.2 yrs	Most defects occur to bearings, blading
22.5 - 30	Condensers	13	338	13,053	1/83.7 yrs	Most defects occur to tubing
22.5 - 30	Boilers	53	149	5,143	1/8.1 yrs	Most defects occur to tubing and valves
30 - Plus	Steam Turbines	43	84	3,188	1/8.3 yrs	Most defects occur to blading
30 - Plus	Condensers	2	84	3,188	1/132.8 yrs	Most defects occur to tubing
30 - Plus	Boilers	0	28	832	-	-

However, there are many factors which suggest that the waste heat boiler/steam turbine system might achieve reliability greater than that indicated in Table 4. These factors include:

- a clean gas turbine exhaust resulting from distillate fuel usage by the gas turbine (in the case of an icebreaking LNG carrier, natural gas might comprise much of the gas turbine fuel),

- convection heat rather than radiant heat transfer in the boiler which provides improved distribution of metal temperatures,

- conservative steam temperatures and pressures which do not require special materials or operating procedures,

- elimination of many of the extenuating factors such as extended ship lay-up, which may have caused certain failures cited in the Lloyd's Register survey.

Although it is difficult to apply the narration of this chapter towards an estimation of the reliability to be achieved by a combined cycle plant installed on a Canadian Arctic icebreaker, the land-based and marine operating experience of gas turbines, boilers and steam turbines does suggest optimism for high availability. Investigations conducted in the course of this study did not identify any elements of the system which might be prone to failure aboard an icebreaking vessel.

At this juncture, it is appropriate to describe the Unites States' Navy Rankine Cycle Energy Recovery (RACER) programme.

4.4 The RACER Programme

The U. S. Navy RACER programme takes its name from the energy recovery provided by a rankine cycle boiler/steam turbine system installed on the exhaust of an aero-derived gas turbine. The Navy initiated the programme to develop a high efficiency cruise propulsion system which extends the range of their non-nuclear surface combatent fleet and provides fuel economy. The programme entails development of a waste heat boiler/steam turbine system to accompany a general Electric LM 2500 aero-derived gas turbine.

In 1979, the Navy engaged General Electric, Solar Turbines International and Garrett Corporation to develop a design of the boiler/steam system of the combined cycle plant. This system design must achieve certain objectives including:

- mean time between failure of over 10,000 hours,

- minimum of shipboard maintenance, approximately 6 manhours per week average,

- fully automatic control.

These objectives are testimony that the U. S. Navy feels extremely high reliability may be achieved by the combined cycle plant. The fact that all three firms submitted design proposals, substantiated by supplementary confirmation studies, which in theory satisfies the U. S. Navy's objectives, indicates suppliers, manufacturers and designers also feel that high reliability may be achieved by the combined cycle plant. Discussions with the three companies involved in the programmes indicate that this high reliability is achieved through:

- minimizing the need for make-up water reducing potential for internal corrosion,

- selection of materials to resist corrosion, erosion and fatigue,

- simplifying the control system.

In late 1980, the three companies submitted their proposals. The U. S. Navy has evaluated these designs and selected the Solar Turbines proposal for proto-type development. This proto-type unit will be thoroughly tested in a land-based environment during the period 1982-84. The U. S. Navy is hopeful to have a combined cycle system in operation by 1986. Figure 23 further describes the proposed schedule of the RACER development programme.

4.5 Potential Improvements of the COGAS System

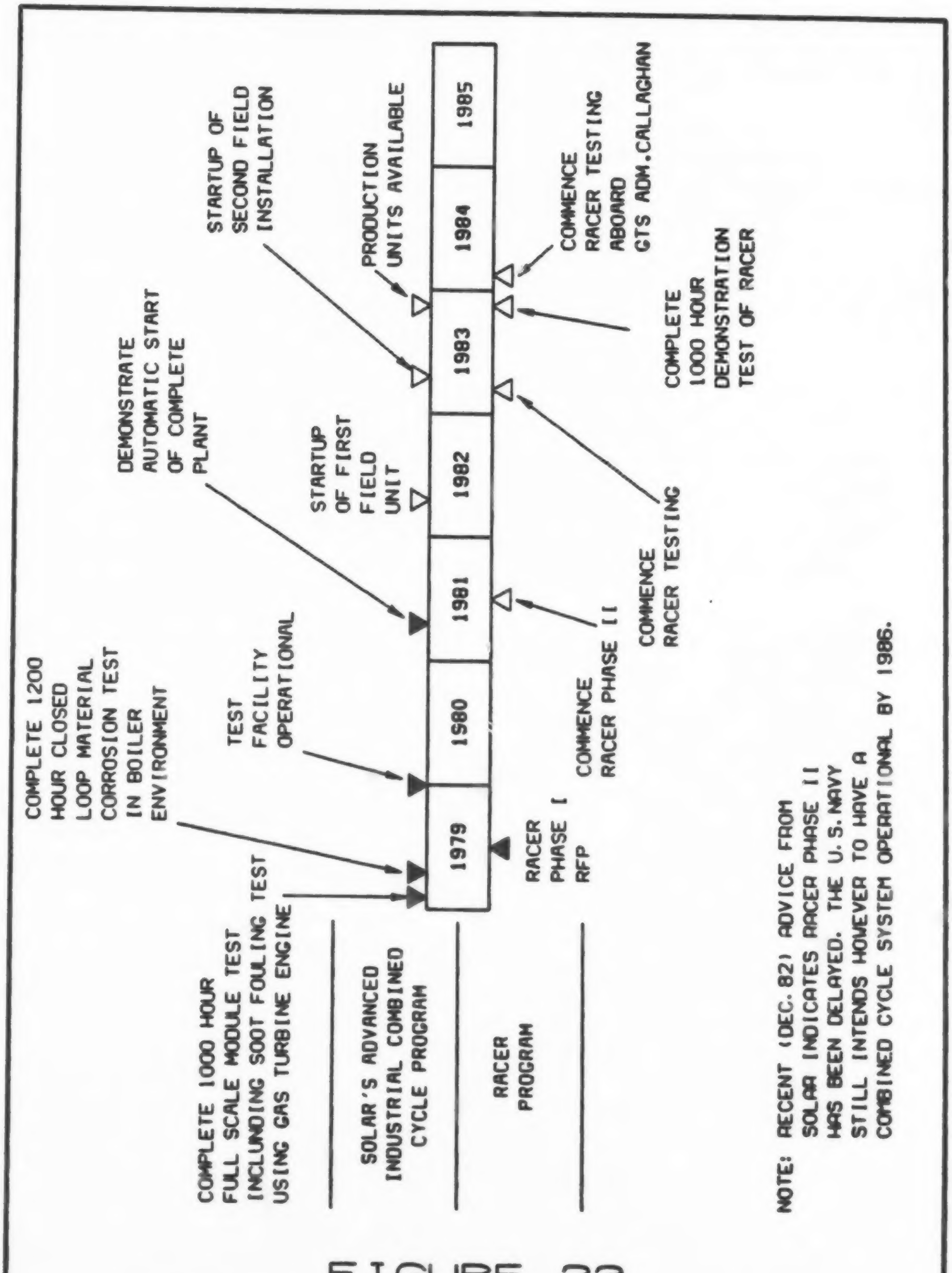
Discussions with manufacturers of boilers, steam turbines, condensers and associated equipment revealed that potential design improvements to the steam portion of the combined cycle system are now under study. These potential design improvements include:

- modification of controls to increase simplicity, efficiency, and reliability,

- selection of materials to increase reliability; materials under evaluation include Incoloy 800 for boiler tubing, titanium blades for steam turbines and titanium sheeting for condensers,

- boiler tubing design to minimize noise and improve efficiency,

- selection of components to reduce the need to pre-treat water, and minimize system leakage and subsequent make-up water requirements.


1979
1980
1981
1982
1983
1984
1985
COMPLETE 1000 HOUR FULL SCALE MODULE TEST INCLUNDING SOOT FOULING TEST USING GAS TURBINE ENGINE
SOLAR'S ADVANCED INDUSTRIAL COMBINED CYCLE PROGRAM
RACER PROGRAM
COMPLETE 1200 HOUR CLOSED LOOP MATERIAL CORROSION TEST IN BOILER ENVIRONMENT
TEST FACILITY OPERATIONAL
DEMONSTRATE AUTOMATIC START OF COMPLETE PLANT
STARTUP OF FIRST FIELD UNIT
STARTUP OF SECOND FIELD INSTALLATION
PRODUCTION UNITS AVAILABLE
RACER PHASE I RFP
COMMENCE RACER PHASE I
COMMENCE RACER PHASE II
COMMENCE RACER TESTING
COMPLETE 1000 HOUR DEMONSTRATION TEST OF RACER
COMMENCE RACER TESTING ABOARD GTS ADM. CALLAGHAN

NOTE: RECENT (DEC. 82) ADVICE FROM SOLAR INDICATES RACER PHASE II HAS BEEN DELAYED. THE U.S. NAVY STILL INTENDS HOWEVER TO HAVE A COMBINED CYCLE SYSTEM OPERATIONAL BY 1986.

FIGURE 23
PROPOSED SCHEDULE OF UNITED STATES NAVY RACER DEVELOPMENT PROGRAMME

These features, plus others, will render the combined cycle system more reliable, more efficient and easier to operate in the future.

5. CONCLUSIONS

Previous studies, including a report prepared for Transport Canada in 1978 and 1979 by Pratt and Whitney Canada Ltd., concluded that the installation of a combined gas turbine and steam turbine propulsion system in a marine application might offer economic benefit. This is particularly true for a liquid natural gas carrier where the boil-off, occurring naturally from cargo, provides an ideal source of fuel for the gas turbines.

However, with limited exception, combined cycle propulsion is untested and untried in a marine application. As a result, there remains a considerable uncertainty surrounding the technical suitability of a combined cycle system in a marine application, particularly an icebreaker where manoeuvers will cause numerous fluctuations in power. This study addresses the technical suitability of a combined cycle propulsion system in a Canadian Arctic icebreaking application and particularly:

- the ability of the system to withstand the icebreaking duties

- the ability of the system to deliver maximum power when required

- the ability of the system to perform reliably in an icebreaking application.

This study concludes that:

- a combined gas turbine and steam turbine propulsion system may be designed to withstand the rigors and unusual ambient conditions associated with operating in the Canadian Arctic

- preliminary investigation of combined cycle system response during manoeuvers associated with icebreaking suggests this propulsion system is technically suitable for a Canadian Arctic icebreaker.

Additionally, this study concludes:

- the Rolls Royce RB 211 and General Electric LM 2500 aero-derived gas turbines appear suitable for a Canadian Arctic icebreaker as a result of their ability to accept rapid load changes.

- a waste heat boiler, suitable for Canadian Arctic icebreaking application, might have the following characteristics:

  - an approach temperature of 44°C (80°F) which represents a reasonable compromise between steam cycle efficiency and boiler capital cost.

  - a pinch point of 22°C (40°F) which also represents a reasonable compromise between efficiency and capital cost.

  - a minimum exhaust gas stack temperature of 126.7°C (260°F) to avoid the sulphur dew point and potential corrosion of boiler tubing.

  - superheated steam conditions at boiler outlet of approximately 400°C (752°F) temperature and 2.53 MN/m^2 (25 atmosphere) pressure.

  - single pressure boiler to ease control complexity.

  - no supplementary firing.

  - capability of dry firing.

  - one boiler for each gas turbine exhaust.

- a steam turbine, suitable for Canadian Arctic icebreaking application might be designed with objectives of simplicity and reliability with particular attention towards accepting "wet" steam which may be produced during operation in low ambient temperatures.

- a constant pressure system might be selected as a method for controlling the steam plant since this provides automatic, reliable and proven control.

- evaluation of preliminary dynamic analysis of the combined cycle system undertaking icebreaking manoeuvers suggests that this system is technically feasible for Canadian Arctic icebreaking application.

- a simulation, conducted by Solar Turbines International, of a General Electric LM 2500 gas turbine equipped with Solar's "once through" dual pressure boiler on the exhaust stack, demonstrated that during the ice ramming cycle, the steam system does not experience significant fluctuation; this suggests that the combined cycle system will not experience unacceptable thermal cycle fatigue and will be able to satisfy power demands in an icebreaking application.

- evaluation of previously published results of dynamic anlayses of combined cycle propulsion having a conventional single pressure waste heat boiler equipped with a steam drum, suggests that the available steam turbine power does not fluctuate rapidly with changes in gas turbine power and that the combined cycle system is stable, and thus able to satisfy power demands during icebreaking.

- review of land-based experience of combined cycle systems and marine experience of high pressure boilers, steam turbines and condensers suggests that a combined cycle system in a Canadian Arctic icebreaking application should achieve high availability.

It must be emphasized that, although this study concludes that a combined cycle propulsion system appears technically suitable for installation in a Canadian Arctic icebreaker, more detailed investigation of the plant must be conducted to gauge with confidence COGAS response during icebreaking manoeuvers.

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3

TECHNICAL EVALUATION OF COMBINED GAS TURBINE AND STEAM TURBINE PROPULSION SYSTEM FOR CANADIAN ARCTIC ICEBREAKING DUTY

TP 4515E

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APPENDICES

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For

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17. Abstract

This study addresses the technical suitability, for installation aboard a Canadian Arctic icebreaking vessel, of a propulsion system consisting of aero-derived gas turbines, waste heat boilers and steam turbines. In particular, the study addresses the ability of this propulsion system to:

- withstand rigorous icebreaking duties;
- delivery maximum power when required;
- perform reliably in an icebreaking application.

This study develops a preliminary design for combined cycle system suitable for an Arctic icebreaker. The study evaluates the ability and performance of this system in conducting icebreaking manoeuvres by analyzing the results of computer-based simulations. The study assess the reliability of combined cycle propulsion for an icebreaking vessel through evaluation of experience of land-based combined cycle systems, marine waste heat boiler systems and marine high pressure steam propulsion systems. The study concludes that a combined cycle propulsion system appears to be technically suitable for installation in a Canadian Arctic icebreaking vessel.

18. Key Words

Icebreaking, Ice Class, Ships, Combined Cycle Propulsion, Gas, Steam Turbines, Waste Heat Boilers

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I.M. Bayly

17. Résumé

Ce rapport traite de l'étude de faisabilité concernant la mise en place à bord d'un navire brise-glace canadien d'un système propulsif comprenant des turbines à gaz dérivées des turbomachines pour avions, des chaudières de récupération et des turbines à vapeur, L'étude a porté plus particulièrement sur l'aptitude d'un tel système à:

- résister aux conditions rigoureuses de service
- fournir au moment voulu la puissance maximale exigée
- fonctionner fiablement dans les opérations de déglaçage.

L'étude a dégagé le concept d'un système propulsif combinant les éléments précités et convenant aux opérations dans l'Arctique, concept dont elle évalue les caractéristiques et les performances par l'examen du résultat de simulations faites à l'aide d'un ordinateur; elle en évalue aussi la fiabilité dans le contexte d'un navire brise-glace, à la lumière de l'expérience acquise dans les applications terrestres des systèmes propulsifs de cette nature et dans l'exploitation de systèmes de propulsion navale à chaudières de récupération et turbines à vapeur haute pression. Enfin, l'étude conclut qu'un système propulsif du type proposé serait techniquement envisageable pour un navire brise-glace de cote arctique opérant dans les eaux canadiennes.

18. Mots-clés
Navire brise-glace, navire de cote arctique, cycle propulsif, turbines à gaz et à vapeur, chaudières de récupération

19. Diffusion
Les organismes parraines dispose d'un nombre limité d'exemplaires

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LIST OF APPENDICES

Appendix 1 . . . . Survey of Incidents Recorded by Lloyd's Register of Shipping for Modern Marine Steam Turbine Propulsion Plants.

Appendix 2 . . . . Materials and Correspondence Received During the Course of the Study. (Catalogue of materials is provided at the beginning of Appendix 2)

APPENDIX 1

Survey of Incidents Recorded by Lloyd's Register of Shipping for Modern Marine Steam Turbine Propulsion Plants

```
QUERY HOLDFLS                                                                                          PAGE      1
MAIN W.T.MEMBRANE WALL BOILERS FITTED TO SHIPS BUILT TO LR CLASS BETWEEN JANUARY 1960 AND DECEMBER 1979 . DEVELOPING 20000 HP AND
OVER AND 40 BAR MINIMUM STEAM PRESSURE      INV 1507

          SHP RANGE         TIME INTERVAL      NO. OF    NO. AT     MONTHS      INCIDENCE
                                               DEFECTS     RISK     SERVICE        RATE

          20000 - 29999     7001      7512       12        13         800          1.50
          20000 - 29999     7601      7912       11        12         366          3.01
          30000 - 39999     7001      7512       25        63        1638          1.59
          30000 - 39999     7601      7912       35        86        3505          1.00
          40000 - 49999     7001      7512        0         8          48          0.00
          40000 - 49999     7601      7912        1        28         832          0.12
```

1.18 overall.

```
      N.B.     1) A DEFECT IS COUNTED ONCE AS AN INCIDENT REPORTED AT A POINT IN TIME
                  REGARDLESS OF THE EXTENT OR SEVERITY
               2) SERVICE TIME IS CALENDER TIME EXCLUDING LAY-UP TIME
               3) INCIDENCE RATE IS THE RATIO OF NUMBER OF DEFECTS TO THE AGGREGATE
                  SERVICE TIMES 100
               4) NO. AT RISK ARE MAIN BOILERS IN PAIRS OR AS A SINGLE WITH A "GET
                  YOU HOME" . WHERE THERE IS ONE MAIN STEAM TURBINE PER SHIP
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QUERY HOLDFL3                                                                                          PAGE    1
MAIN W.T.MEMBRANE WALL BOILERS FITTED TO SHIPS BUILT TO LR CLASS BETWEEN JANUARY 1960 AND DECEMBER 1979 . DEVELOPING 20000 HP AND
OVER AND 40 BAR MINIMUM STEAM PRESSURE      INV 1507

SHP RANGE        TIME INTERVAL   DEFECTIVE LOCATION                          DEFECT MODE                  REPAIR                       COUNT

20000 - 29999   7001     7512   BRICKWORK                                   FAILED, DEFECTIVE            UNSPEC. REPAIR ✓               1
20000 - 29999   7001     7512   CASING                                      CHEM. ACTION                 LOCATION RENEWED ✓             1
20000 - 29999   7001     7512   INLET PIPE (DESUPERHEATER)                  POROUS, LEAKING, HOLED       LOCATION RENEWED ✓             1
20000 - 29999   7001     7512   SUPERHEATER SUPPORTS                        BROKEN, SHEARED              LOCATION RENEWED ✓             2
20000 - 29999   7001     7512   SUPERHEATER SUPPORTS                        CHEM. ACTION                 LOCATION RENEWED ✓             1
20000 - 29999   7001     7512   SUPERHEATER SUPPORTS                        CRACKED                      WELDED ✓                       1
20000 - 29999   7001     7512   SUPERHEATER SUPPORTS                        FAILED, DEFECTIVE            LOCATION RENEWED ✓             1
20000 - 29999   7001     7512   TUBES & ELEMENTS (SUPERHEATER)              BROKEN, SHEARED              DEFERRED ✓                     1
20000 - 29999   7001     7512   TUBES & ELEMENTS (SUPERHEATER)              BURST, RUPTURED              LOCATION RENEWED ✓             1
20000 - 29999   7001     7512   TUBES & ELEMENTS (SUPERHEATER)              MISALIGNED, DEFORMED         LOCATION RENEWED ✓             1
20000 - 29999   7001     7512   TUBES & ELEMENTS (SUPERHEATER)              MISALIGNED, DEFORMED         UNSPEC. REPAIR ✓               1
20000 - 29999   7001     7512   TUBES & ELEMENTS (SUPERHEATER)              POROUS, LEAKING, HOLED       WELDED ✓                       1
20000 - 29999   7001     7512   TUBES,MISC. (SPEC.)(GENERATING)             BURNT/SQUEEZED/HAMMERED      LOCATION RENEWED ✓             1
20000 - 29999   7001     7512   TUBES,MISC. (SPEC.)(GENERATING)             MECH. ACTION                 LOCATION RENEWED ✓             1
20000 - 29999   7001     7512   TUBES,MISC. (SPEC.)(GENERATING)             MECH. ACTION                 WELDED ✓                       1
20000 - 29999   7001     7512   WALL TUBES (MEMBRANE)                       BURNT/SQUEEZED/HAMMERED      LOCATION RENEWED               1
20000 - 29999   7601     7912   MEMBRANE WALLS                              .CRACKED                     LOCATION RENEWED ✓             1
20000 - 29999   7601     7912   ROOF TUBES (MEMBRANE)                       CRACKED                      LOCATION RENEWED ✓             1
20000 - 29999   7601     7912   SAFETY VALVES                               POROUS, LEAKING, HOLED       LOCATION RENEWED ✓             1
20000 - 29999   7601     7912   SUPERHEATER SUPPORTS                        BROKEN, SHEARED              LOCATION RENEWED ✓             1
20000 - 29999   7601     7912   SUPERHEATER SUPPORTS                        BURNT/SQUEEZED/HAMMERED      LOCATION RENEWED ✓             1
20000 - 29999   7601     7912   SUPERHEATER SUPPORTS                        CRACKED                      LOCATION RENEWED ✓             1
20000 - 29999   7601     7912   SUPERHEATER SUPPORTS                        LOOSE, TWISTED, SLIPPED      LOCATION RENEWED .             1
20000 - 29999   7601     7912   TUBE SUPPORTS (FOR GENERATING TUBES)        CRACKED                      WELDED ✓                       1
```

```
QUERY HOLDFL3
                                                                                                        PAGE      2
20000 - 29999  7601    7912  TUBES & ELEMENTS (SUPERHEATER) BROKEN, SHEARED           LOCATION RENEWED ✓          1
20000 - 29999  7601    7912  TUBES & ELEMENTS (SUPERHEATER) CHEM. ACTION              LOCATION RENEWED /          2
20000 - 29999  7601    7912  TUBES & ELEMENTS (SUPERHEATER) FLAKING,STRIPPED,TORN     TEMPORARY REPAIR ✓          1
20000 - 29999  7601    7912  TUBES & ELEMENTS (SUPERHEATER) MISALIGNED, DEFORMED      LOCATION RENEWED X          2
20000 - 29999  7601    7912  TUBES & ELEMENTS (SUPERHEATER) POROUS, LEAKING, HOLED    LOCATION RENEWED ✓          2
20000 - 29999  7601    7912  TUBES & ELEMENTS (SUPERHEATER) POROUS, LEAKING, HOLED    TEMPORARY REPAIR X          1
20000 - 29999  7601    7912  TUBES,MISC. (SPEC.) (GENERATING) MISALIGNED, DEFORMED    DEFERRED ✓                  1
20000 - 29999  7601    7912  WALL TUBES (MEMBRANE)          BURNT/SQUEEZED/HAMMERED   WELDED                      1
20000 - 29999  7601    7912  WALL TUBES (MEMBRANE)          CRACKED                   WE_DED ✓                    1
20000 - 29999  7601    7912  WALL TUBES (MEMBRANE)          SCALED,CHOKED             WE_DED X                    1
30000 - 39999  7001    7512  CASING                         BURST, RUPTURED           LOCATION RENEWED ✓          1
30000 - 39999  7001    7512  CHECK VALVES                   CRACKED                   COMPONENT RENEWED ✓         1
30000 - 39999  7001    7512  CHECK VALVES                   POROUS, LEAKING, HOLED    COMPONENT RENEWED ✓         1
30000 - 39999  7001    7512  HEADERS (ECONOMISER)           CRACKED                   PT. RENWD-MOD. DESGN/MODIFIED ✓  1
30000 - 39999  7001    7512  I.W.O. MANHOLE (DRUM)          MECH. ACTION              MACHINED, POLISHED, HONED /  1
30000 - 39999  7001    7512  I.W.O. MOUNTINGS & MTG.PADS (DRUM) POROUS, LEAKING, HOLED  MISC. X                   1
30000 - 39999  7001    7512  INLET PIPE (DESUPERHEATER)     BURNT/SQUEEZED/HAMMERED   WE_DED ✓                    1
30000 - 39999  7001    7512  INLET PIPE (DESUPERHEATER)     CHEM. ACTION              WELDED X                    1
30000 - 39999  7001    7512  MISC. (SPEC.) (SUPERHEATER)    POROUS, LEAKING, HOLED    DEFERRED X                  1
30000 - 39999  7001    7512  MISC. WELD ATTCHMNTS. (DRUM)   CRACKLD                   MACHINED, POLISHED, HONED X  2
30000 - 39999  7001    7512  ROOF TUBES (MEMBRANE)          CRACKED                   WE_DED ✓                    1
30000 - 39999  7001    7512  SAFETY VALVES                  CHEM. ACTION              LOCATION RENEWED X          1
30000 - 39999  7001    7512  SAFETY VALVES                  MISALIGNED, DEFORMED      DEFERRED X                  1
30000 - 39999  7001    7512  SAFETY VALVES                  POROUS, LEAKING, HOLED    LOCATION RENEWED ✓          1
30000 - 39999  7001    7512  SAFETY VALVES                  POROUS, LEAKING, HOLED    MISC. ✓                     1
30000 - 39999  7001    7512  SAFETY VALVES                  POROUS, LEAKING, HOLED    UNSPEC. REPAIR X            1
30000 - 39999  7001    7512  STOP VALVES                    CRACKED                   COMPONENT RENEWED           1
30000 - 39999  7001    7512  SUPERHEATER SUPPORTS           CHEM. ACTION              DEFERRED /                  1
30000 - 39999  7001    7512  SUPERHEATER SUPPORTS           FAILED, DEFECTIVE         LOCATION RENEWED            1
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QUERY HOLDFL3                                                                                                  PAGE     3
30000 - 39999  7001    7512  TUHEPLATE                              CHEM. ACTION              LOCATION RENEWED          1
30000 - 39999  7001    7512  TUBES (AIR HEATER)                     CHEM. ACTION              LOCATION RENEWED ✓        1
30000 - 39999  7001    7512  TUBES (AIR HEATER)                     CRACKED                   LOCATION RENEWED ✓        1
30000 - 39999  7001    7512  TUBES (AIR HEATER)                     FAILED. DEFECTIVE         WELDED ✓                  1
30000 - 39999  7001    7512  TUBES & ELEMENTS (SUPERHEATER)         BURST. RUPTURED           TEMPORARY REPAIR ✓        1
30000 - 39999  7001    7512  TUBES & ELEMENTS (SUPERHEATER)         LOOSE. TWISTED. SLIPPED   LOCATION RENEWED          1
30000 - 39999  7001    7512  TUBES & ELEMENTS (SUPERHEATER)         MISALIGNED. DEFORMED      LOCATION RENEWED          1
30000 - 39999  7001    7512  TUBES.MISC. (SPEC.)(GENERATING)        BURNT/SQUEEZED/HAMMERED   LOCATION RENEWED ✓        1
30000 - 39999  7001    7512  TUBES.MISC. (SPEC.) (GENERATING)       BURNT/SQUEEZED/HAMMERED   WELDED ✓                  1
30000 - 39999  7001    7512  VALVES                                 CRACKED                   MACHINED. POLISHED. HONED ✓  1
30000 - 39999  7001    7512  VALVES                                 CRACKED                   WELDED ✓                  1
30000 - 39999  7001    7512  WALL TUBES (MEMBRANE)                  CHEM. ACTION              WELDED ✓                  1
30000 - 39999  7001    7512  WALL TUBES (MEMBRANE)                  MECH. ACTION              WELDED ✓                  1
30000 - 39999  7001    7512  WALL TUBES (MEMBRANE)                  MISALIGNED. DEFORMED      LOCATION RENEWED          1
30000 - 39999  7001    7512  WALL TUBES (MEMBRANE)                  POROUS. LEAKING. HOLED    WELDED ✓                  1
30000 - 39999  7601    7912  CHECK VALVES                           CRACKED                   WELDED ✓                  1
30000 - 39999  7601    7912  DOWNCOMER PIPES                        CHEM. ACTION              DEFERRED ✓                1
30000 - 39999  7601    7912  DOWNCOMER PIPES                        MECH. ACTION              DEFERRED ✓                1
30000 - 39999  7601    7912  ELSE SPEC (ECONOMISER)                 SCALED.CHOKED             UNSPEC. REPAIR ✓          1
30000 - 39999  7601    7912  FEED SYSTEM (& BLR. CONTA4.)           FAILED. DEFECTIVE         MISC. ✓                   1
30000 - 39999  7601    7912  FURNACE MISC. (SPEC.)                  CRACKED                   UNSPEC. REPAIR ✓          1
30000 - 39999  7601    7912  HEADERS (ECONOMISER)                   MECH. ACTION              WELDED ✓                  1
30000 - 39999  7601    7912  HEADERS (ECONOMISER)                   POROUS. LEAKING. HOLED    WELDED ✓                  2
30000 - 39999  7601    7912  MEMBRANE WALLS                         BURNT/SQUEEZED/HAMMERED   LOCATION RENEWED ✓        1
30000 - 39999  7601    7912  MEMHRANE WALLS                         CHEM. ACTION              LOCATION RENEWED ✓        1
30000 - 39999  7601    7912  MEMBRANE WALLS                         POROUS. LEAKING. HOLED    LOCATION RENEWED ✓        1
30000 - 39999  7601    7912  MISC. (SPEC.) (SUPERHEATER)            CHEM. ACTION              DEFERRED ✓                1
30000 - 39999  7601    7912  MISC. (SPEC.) (SUPERHEATER)            FAILED. DEFECTIVE         LOCATION RENEWED ✓        1
30000 - 39999  7601    7912  MISC. WELD ATTCHMNTS. (W.W.)           CRACKED                   MACHINED. POLISHED. HONED ✓  1
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QUERY HOLDFL3
                                                                                                                  PAGE
30000 - 39999  7601   7912  PIPING (FEED SYSTEM)              CHEM. ACTION                LOCATION RENEWED          1
30000 - 39999  7601   7912  PIPING (FEED SYSTEM)              CRACKED                     WELDED /                  1
30000 - 39999  7601   7912  PIPING (FEED SYSTEM)              POROUS. LEAKING. HOLED      LOCATION RENEWED /        1
30000 - 39999  7601   7912  SAFETY VALVES                     CRACKED                     LOCATION RENEWED          1
30000 - 39999  7601   7912  SAFETY VALVES                     FAILED. DEFECTIVE           UNSPEC. REPAIR            1
30000 - 39999  7601   7912  SAFETY VALVES                     POROUS. LEAKING. HOLED      UNSPEC. REPAIR            2
30000 - 39999  7601   7912  SEAM (WELD) (CASING)              MECH. ACTION                DEFERRED /                1
30000 - 39999  7601   7912  STOP VALVES                       CRACKED                     WELDED /                  2
30000 - 39999  7601   7912  STOP VALVES                       FAILED. DEFECTIVE           LOCATION RENEWED /        1
30000 - 39999  7601   7912  STUB PIPES                        CRACKED                     TEMPORARY REPAIR /        1
30000 - 39999  7601   7912  STUB PIPES                        CRACKED                     WELDED /                  3
30000 - 39999  7601   7912  TUBES (ECONOMISER)                CRACKED                     LOCATION RENEWED /        1
30000 - 39999  7601   7912  TUBES (AIR HEATER)                POROUS. LEAKING. HOLED      COMPONENT RENEWED /       2
30000 - 39999  7601   7912  TUBES (ECONOMISER)                POROUS. LEAKING. HOLED      LOCATION RENEWED.         1
30000 - 39999  7601   7912  TUBES & ELEMENTS (SUPERHEATER)    CHEM. ACTION                LOCATION RENEWED /        1
30000 - 39999  7601   7912  TUBES & ELEMENTS (SUPERHEATER)    CRACKED                     LOCATION RENEWED          1
30000 - 39999  7601   7912  TUBES & ELEMENTS (SUPERHEATER)    CRACKED                     TEMPORARY REPAIR /        1
30000 - 39999  7601   7912  TUBES & ELEMENTS (SUPERHEATER)    CRACKED                     WELDED /                  1
30000 - 39999  7601   7912  TUBES & ELEMENTS (SUPERHEATER)    MISALIGNED. DEFORMED        UNSPEC. REPAIR /          1
30000 - 39999  7601   7912  TUBES & ELEMENTS (SUPERHEATER)    MISALIGNED. DEFORMED        WELDED X                  1
30000 - 39999  7601   7912  TUBES & ELEMENTS (SUPERHEATER)    POROUS. LEAKING. HOLED      LOCATION RENEWED /        1
30000 - 39999  7601   7912  TUBES.MISC. (SPEC.)(GENERATING)   FAILED. DEFECTIVE           LOCATION RENEWED /        1
30000 - 39999  7601   7912  TUBES.MISC. (SPEC.)(GENERATING)   MISALIGNED. DEFORMED        LOCATION RENEWED /        1
30000 - 39999  7601   7912  TUBES.MISC. (SPEC.)(GENERATING)   POROUS. LEAKING. HOLED      LOCATION RENEWED /        1
30000 - 39999  7601   7912  VENT PIPES                        CRACKED                     WELDED                    1
30000 - 39999  7601   7912  WALL TUBES (MEMBRANE)             BURST. RUPTURED             LOCATION RENEWED          1
30000 - 39999  7601   7912  WALL TUBES (MEMBRANE)             CRACKED                     LOCATION RENEWED /        4
30000 - 39999  7601   7912  WALL TUBES (MEMBRANE              MISALIGNED. DEFORMED        LOCATION RENEWED /        2
40000 - 49999  7601   7912  FEED SYSTEM (& BLR. CONTAM.)      FAILED. DEFECTIVE           CHEM. TRTMENT. DEGREASED /  1
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N.B.    THE DEFECT COUNT IS GREATER THAN BEFORE DUE TO COUNTING BY EXTENT
        WITHIN THE INCIDENT
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20000 - 29999  7601   7912  JOURNAL BRNGS. HOUSING (ROTOR)  MECH. ACTION                LOCATION RENEWED ✓                1
20000 - 29999  7601   7912  JOURNAL BRNGS. LINING (ROTOR)   BURNT/SQUEEZED/HAMMERED     LOCATION RENEWED ✓                4
20000 - 29999  7601   7912  JOURNAL BRNGS. LINING (ROTOR)   BURNT/SQUEEZED/HAMMERED     UNSPEC. REPAIR ✓                  2
20000 - 29999  7601   7912  JOURNAL BRNGS. LINING (ROTOR)   FAILED, DEFECTIVE           LOCATION RENEWED ✓                1
20000 - 29999  7601   7912  JOURNAL BRNGS. LINING (ROTOR)   MECH. ACTION                MACHINED, POLISHED, HONED ✓       2
20000 - 29999  7601   7912  JOURNAL BRNGS. LINING (ROTOR)   MISALIGNED, DEFORMED        LOCATION RENEWED ✓                1
20000 - 29999  7601   7912  JOURNALS (ROTOR)                MECH. ACTION                LOCATION RENEWED ✓                1
20000 - 29999  7601   7912  JOURNALS (ROTOR)                MECH. ACTION                MACHINED, POLISHED, HONED ✓       1
20000 - 29999  7601   7912  JOURNALS (ROTOR)                MECH. ACTION                UNSPEC. REPAIR ✓                  1
20000 - 29999  7601   7912  LABYRINTH & DUMMY PACKING       FAILED, DEFECTIVE           LOCATION RENEWED ✓                1
20000 - 29999  7601   7912  LABYRINTH & DUMMY PACKING       MECH. ACTION                LOCATION RENEWED ✓                1
20000 - 29999  7601   7912  LABYRINTH & DUMMY PACKING       MISALIGNED, DEFORMED        UNSPEC. REPAIR ✓                  1
20000 - 29999  7601   7912  LABYRINTH & DUMMY PACKING       SEIZED, JAMMED              FAIRED, ADJUSTED, EXPANDED ✓      1
20000 - 29999  7601   7912  MANOEUVRING VALVES              CRACKED                     DEFERRED ✓                        1
20000 - 29999  7601   7912  MANOEUVRING VALVES              CRACKED                     WELDED ✓                          1
20000 - 29999  7601   7912  MANOEUVRING VALVES              FAILED, DEFECTIVE           UNSPEC. REPAIR ✓                  1
20000 - 29999  7601   7912  MANOEUVRING VALVES              MECH. ACTION                LOCATION RENEWED ✓                1
20000 - 29999  7601   7912  MANOEUVRING VALVES              MECH. ACTION                MISC. ✓                           2
20000 - 29999  7601   7912  MISC. (SPEC.) VALVES            BROKEN, SHEARED             COMPONENT RENEWED ✓               1
20000 - 29999  7601   7912  MISC. (SPEC.) VALVES            CRACKED                     TEMPORARY REPAIR ✓                1
20000 - 29999  7601   7912  MISC. (SPEC.) VALVES            LOOSE, TWISTED, SLIPPED     COMPONENT RENEWED ✓               1
20000 - 29999  7601   7912  MISC. (SPEC.) VALVES            MECH. ACTION                MISC. ✓                           1
20000 - 29999  7601   7912  MISC. (SPEC.) VALVES            MISALIGNED, DEFORMED        LOCATION RENEWED ✓                1
20000 - 29999  7601   7912  MISC. (SPEC.) VALVES            MISALIGNED, DEFORMED        MISC. ✓                           1
20000 - 29999  7601   7912  NOZZLE BOX                      MECH. ACTION                MACHINED, POLISHED, HONED ✓       1
20000 - 29999  7601   7912  NOZZLE PLATE                    CRACKED                     LOCATION RENEWED ✓                1
20000 - 29999  7601   7912  NOZZLE PLATE                    POROUS, LEAKING, HOLED      LOCATION RENEWED ✓                1
20000 - 29999  7601   7912  ROTOR BLADING                   BROKEN, SHEARED             DEFERRED ✓                        1
20000 - 29999  7601   7912  ROTOR BLADING                   CHEM. ACTION                DEFERRED ✓                        1
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QUERY HANDLED

20000 - 29999  7601    7912  ROTOR BLADING                      CRACKED                    MACHINED, POLISHED, HONED      1
20000 - 29999  7601    7912  ROTOR BLADING                      FAILED, DEFECTIVE          LOCATION RENEWED               1
20000 - 29999  7601    7912  ROTOR BLADING                      FAILED, DEFECTIVE          MACHINED, POLISHED, HONED      1
20000 - 29999  7601    7912  ROTOR BLADING                      FAILED, DEFECTIVE          MISC.                          2
20000 - 29999  7601    7912  ROTOR BLADING                      LOOSE, TWISTED, SLIPPED    LOCATION RENEWED               2
20000 - 29999  7601    7912  ROTOR BLADING                      MECH. ACTION               DEFERRED                       2
20000 - 29999  7601    7912  ROTOR BLADING                      MECH. ACTION               TEMPORARY REPAIR ✓             1
20000 - 29999  7601    7912  ROTOR BLADING                      MISALIGNED, DEFORMED       COMPONENT RENEWED              1
20000 - 29999  7601    7912  ROTOR BLADING                      MISALIGNED, DEFORMED       MACHINED, POLISHED, HONED      1
20000 - 29999  7601    7912  ROTOR BLADING                      SCALED,CHOKED              MISC.                          1
20000 - 29999  7601    7912  ROTOR DISCS                        MECH. ACTION               MACHINED, POLISHED, HONED      1
20000 - 29999  7601    7912  ROTOR DISCS                        MECH. ACTION               MISC. ✓                        1
20000 - 29999  7601    7912  SHELL (CONDENSER)                  CHEM. ACTION               WELDED ✓                       1
20000 - 29999  7601    7912  SLIDING FEET (TURBINE CASING)      CHEM. ACTION               MACHINED, POLISHED, HONED      1
20000 - 29999  7601    7912  SPEEDER MOTOR (GOVERNOR)           FAILED, DEFECTIVE          COMPONENT RENEWED              1
20000 - 29999  7601    7912  STAYS (CONDENSER)                  POROUS, LEAKING, HOLED     DEFERRED                       1
20000 - 29999  7601    7912  THRUST BRNGS. (ROTOR)              BURNT/SQUEEZED/HAMMERED    LOCATION RENEWED               1
20000 - 29999  7601    7912  THRUST BRNGS. (ROTOR)              CRACKED                    LOCATION RENEWED               1
20000 - 29999  7601    7912  THRUST BRNGS. (ROTOR)              MECH. ACTION               UNSPEC. REPAIR                 1
20000 - 29999  7601    7912  THRUST COLLAR (ROTOR)              CRACKED                    LOCATION RENEWED               1
20000 - 29999  7601    7912  TUBEPLATE (CONDENSER)              FAILED, DEFECTIVE          LOCATION RENEWED               1
20000 - 29999  7601    7912  TUBES (CONDENSER)                  CHEM. ACTION               LOCATION RENEWED               1
20000 - 29999  7601    7912  TUBES (CONDENSER)                  CRACKED                    LOCATION RENEWED ✓             1
20000 - 29999  7601    7912  TUBES (CONDENSER)                  FAILED, DEFECTIVE          LOCATION RENEWED ✓             1
20000 - 29999  7601    7912  TUBES (CONDENSER)                  MISALIGNED, DEFORMED       UNSPEC. REPAIR                 1
20000 - 29999  7601    7912  TUBES (CONDENSER)                  POROUS, LEAKING, HOLED     DEFERRED                       1
20000 - 29999  7601    7912  TUBES (CONDENSER)                  POROUS, LEAKING, HOLED     LOCATION RENEWED               1
20000 - 29999  7601    7912  UNSPEC (CONDENSER)                 FAILED, DEFECTIVE          COMPONENT RENEWED              1
20000 - 29999  7601    7912  UNSPEC (CONDENSER)                 FAILED, DEFECTIVE          LOCATION RENEWED ✓             1
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QUERY NUMBER 1
                                                                                                        PAGE    8
20000 - 29999  7001    7912  UNSPEC (CONDENSER)                     MISALIGNED, DEFORMED      COMPONENT RENEWED                 1
20000 - 29999  7001    7912  WATERBOX (CONDENSER)                   BROKEN, SHEARED           LOCATION RENEWED                  1
20000 - 29999  7001    7912  WATERBOX (CONDENSER)                   CHEM. ACTION              TEMPORARY REPAIR                  1
20000 - 29999  7001    7912  WATERBOX (CONDENSER)                   CHEM. ACTION              WELDED                            2
20000 - 29999  7001    7912  WATERBOX (CONDENSER)                   CRACKED                   LOCATION RENEWED                  1
30000 - 39999  7001    7512  BLED STEAM VALVES                      BURNT/SQUEEZED/HAMMERED   MACHINED, POLISHED, HONED         1
30000 - 39999  7001    7512  BLED STEAM VALVES                      CRACKED                   MISC.                             1
30000 - 39999  7001    7512  BLED STEAM VALVES                      CRACKED                   WELDED                            1
30000 - 39999  7001    7512  CASING  BLADING                        BROKEN, SHEARED           WELDED                            1
30000 - 39999  7001    7512  CASING  BLADING                        BURNT/SQUEEZED/HAMMERED   WELDED                            1
30000 - 39999  7001    7512  CASING  BLADING                        CRACKED                   MACHINED, POLISHED, HONED         1
30000 - 39999  7001    7512  CASING  BLADING                        MECH. ACTION              LOCATION RENEWED                  1
30000 - 39999  7001    7512  CASING  BLADING                        MECH. ACTION              PT. RENWD-MOD. DESGN/MODIFIED     1
30000 - 39999  7001    7512  CASING  BLADING                        MECH. ACTION              TEMPORARY REPAIR                  1
30000 - 39999  7001    7512  CASING  BLADING                        MECH. ACTION              WELDED                            1
30000 - 39999  7001    7512  CASING  BLADING                        MISALIGNED, DEFORMED      DEFERRED                          1
30000 - 39999  7001    7512  CASING  BLADING                        SCALED,CHCKED             CHEM. TRTMENT, DEGREASED          1
30000 - 39999  7001    7512  DIAPHRAGMS                             FAILED, DEFECTIVE         LOCATION RENEWED                  1
30000 - 39999  7001    7512  DIAPHRAGMS                             MISALIGNED, DEFORMED      FAIRED, ADJUSTED, EXPANDED        1
30000 - 39999  7001    7512  DIAPHRAGMS                             MISALIGNED, DEFORMED      UNSPEC. REPAIR                    1
30000 - 39999  7001    7512  JOURNAL BRNGS. HOUSING (ROTOR)         MECH. ACTION              UNSPEC. REPAIR                    1
30000 - 39999  7001    7512  JOURNAL BRNGS. HOUSING (ROTOR)         SEIZED, JAMMED            COMPONENT RENEWED                 1
30000 - 39999  7001    7512  JOURNAL BRNGS. LINING (ROTOR)          BROKEN, SHEARED           LOCATION RENEWED                  3
30000 - 39999  7001    7512  JOURNAL BRNGS. LINING (ROTOR)          BURNT/SQUEEZED/HAMMERED   COMPONENT RENEWED                 4
30000 - 39999  7001    7512  JOURNAL BRNGS. LINING (ROTOR)          BURNT/SQUEEZED/HAMMERED   LOCATION RENEWED                  5
30000 - 39999  7001    7512  JOURNAL BRNGS. LINING (ROTOR)          BURNT/SQUEEZED/HAMMERED   PT. RENWD-MOD. DESGN/MODIFIED     1
30000 - 39999  7001    7512  JOURNAL BRNGS. LINING (ROTOR)          LOOSE, TWISTED, SLIPPED   LOCATION RENEWED                  1
30000 - 39999  7001    7512  JOURNAL BRNGS. LINING (ROTOR)          WORN, TAPERED             LOCATION RENEWED                  1
30000 - 39999  7001    7512  JOURNALS (ROTOR)                       CRACKED                   LOCATION RENEWED                  1
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30000 - 39999  7001     7512  JOURNALS (ROTOR)                 FAILED, DEFECTIVE           COMPONENT RENEWED                    1
30000 - 39999  7001     7512  JOURNALS (ROTOR)                 MECH. ACTION                LOCATION RENEWED                     1
30000 - 39999  7001     7512  JOURNALS (ROTOR)                 MECH. ACTION                MACHINED, POLISHED, HONED            1
30000 - 39999  7001     7512  LABYRINTH & DUMMY PACKING        BURNT/SQUEEZED/HAMMERED     MISC.                                1
30000 - 39999  7001     7512  LABYRINTH & DUMMY PACKING        FAILED, DEFECTIVE           MISC.                                1
30000 - 39999  7001     7512  LABYRINTH & DUMMY PACKING        FLAKING,STRIPPED,TORN       LOCATION RENEWED                     1
30000 - 39999  7001     7512  LABYRINTH & DUMMY PACKING        FLAKING,STRIPPED,TORN       MACHINED, POLISHED, HONED            1
30000 - 39999  7001     7512  LABYRINTH & DUMMY PACKING        MISALIGNED, DEFORMED        MACHINED, POLISHED, HONED            2
30000 - 39999  7001     7512  LABYRINTH & DUMMY PACKING        MISALIGNED, DEFORMED        MISC.                                1
30000 - 39999  7001     7512  LUB. OIL PUMP                    SEIZED, JAMMED              DEFERRED                             1
30000 - 39999  7001     7512  MANOEUVRING VALVES               CRACKED                     LOCATION RENEWED                     1
30000 - 39999  7001     7512  MANOEUVRING VALVES               POROUS, LEAKING, HOLED      PT. RENWD-MOD. DESGN/MODIFIED        1
30000 - 39999  7001     7512  MISC. (SPEC.) VALVES             CRACKED                     COMPONENT RENEWED                    1
30000 - 39999  7001     7512  MISC. (SPEC.) VALVES             CRACKED                     MACHINED, POLISHED, HONED            1
30000 - 39999  7001     7512  MISC. (SPEC.) VALVES             CRACKED                     TEMPORARY REPAIR                     1
30000 - 39999  7001     7512  MISC. (SPEC.) VALVES             FAILED, DEFECTIVE           LOCATION RENEWED                     1
30000 - 39999  7001     7512  MISC. (SPEC.) VALVES             FAILED, DEFECTIVE           MISC.                                1
30000 - 39999  7001     7512  MISC. (SPEC.) VALVES             MECH. ACTION                MACHINED, POLISHED, HONED            1
30000 - 39999  7001     7512  MISC. (SPEC.) VALVES             MISALIGNED, DEFORMED        LOCATION RENEWED                     1
30000 - 39999  7001     7512  MISC. (SPEC.) VALVES             POROUS, LEAKING, HOLED      TEMPORARY REPAIR                     1
30000 - 39999  7001     7512  NOZZLE BOX                       CRACKED                     TEMPORARY REPAIR                     1
30000 - 39999  7001     7512  NOZZLE PLATE                     CHEM. ACTION                LOCATION RENEWED                     1
30000 - 39999  7001     7512  NOZZLE PLATE                     FAILED, DEFECTIVE           LOCATION RENEWED                     1
30000 - 39999  7001     7512  NOZZLE PLATE                     FLAKING,STRIPPED,TORN       TEMPORARY REPAIR                     1
30000 - 39999  7001     7512  NOZZLE PLATE                     MISALIGNED, DEFORMED        COMPONENT RENEWED                    1
30000 - 39999  7001     7512  NOZZLE PLATE                     MISALIGNED, DEFORMED        TEMPORARY REPAIR                     1
30000 - 39999  7001     7512  ROTOR BLADING                    BROKEN, SHEARED             LOCATION RENEWED.                    4
30000 - 39999  7001     7512  ROTOR BLADING                    BROKEN, SHEARED             MISC.                                1
30000 - 39999  7001     7512  ROTOR BLADING                    BROKEN, SHEARED             PT. RENWD-MOD. DESGN/MODIFIED        1
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QUERY HULDFL1
                                                                                                  PAGE   10
30000 - 39999, 7001    7512  ROTOR BLADING        BROKEN, SHEARED            TEMPORARY REPAIR                2
30000 - 39999, 7001    7512  ROTOR BLADING        BROKEN, SHEARED            UNSPEC. REPAIR                  1
30000 - 39999, 7001    7512  ROTOR BLADING        CHEM. ACTION               DEFERRED                        1
30000 - 39999, 7001    7512  ROTOR BLADING        CHEM. ACTION               LOCATION RENEWED                1
30000 - 39999, 7001    7512  ROTOR BLADING        CRACKED                    COMPONENT RENEWED               1
30000 - 39999, 7001    7512  ROTOR BLADING        CRACKED                    DEFERRED                        1
30000 - 39999, 7001    7512  ROTOR BLADING        CRACKED                    LOCATION RENEWED                3
30000 - 39999, 7001    7512  ROTOR BLADING        CRACKED                    MACHINED, POLISHED, HONED       1
30000 - 39999, 7001    7512  ROTOR BLADING        CRACKED                    UNSPEC. REPAIR                  2
30000 - 39999, 7001    7512  ROTOR BLADING        FAILED, DEFECTIVE          COMPONENT RENEWED               2
30000 - 39999, 7001    7512  ROTOR BLADING        FAILED, DEFECTIVE          FAIRED, ADJUSTED, EXPANDED      1
30000 - 39999, 7001    7512  ROTOR BLADING        FAILED, DEFECTIVE          LOCATION RENEWED                1
30000 - 39999, 7001    7512  ROTOR BLADING        FAILED, DEFECTIVE          MISC.                           1
30000 - 39999, 7001    7512  ROTOR BLADING        FLAKING,STRIPPED,TORN      COMPONENT RENEWED               1
30000 - 39999, 7001    7512  ROTOR BLADING        LOOSE, TWISTED, SLIPPED    LOCATION RENEWED                1
30000 - 39999, 7001    7512  ROTOR BLADING        MECH. ACTION               COMPONENT RENEWED               1
30000 - 39999, 7001    7512  ROTOR BLADING        MECH. ACTION               FAIRED, ADJUSTED, EXPANDED      1
30000 - 39999, 7001    7512  ROTOR BLADING        MECH. ACTION               LOCATION RENEWED                1
30000 - 39999, 7001    7512  ROTOR BLADING        MECH. ACTION               MACHINED, POLISHED, HONED       1
30000 - 39999, 7001    7512  ROTOR BLADING        MECH. ACTION               TEMPORARY REPAIR                1
30000 - 39999, 7001    7512  ROTOR BLADING        MISALIGNED, DEFORMED       FAIRED, ADJUSTED, EXPANDED      1
30000 - 39999, 7001    7512  ROTOR BLADING        MISALIGNED, DEFORMED       LOCATION RENEWED                1
30000 - 39999, 7001    7512  ROTOR BLADING        MISALIGNED, DEFORMED       MISC.                           1
30000 - 39999, 7001    7512  ROTOR BLADING        MISALIGNED, DEFORMED       PT. RENWD-MOD. DESGN/MODIFIED   1
30000 - 39999, 7001    7512  ROTOR BLADING        MISSING, LOST              TEMPORARY REPAIR                1
30000 - 39999, 7001    7512  ROTOR BLADING        SCALED,CHOKED              CHEM. TRTMENT, DEGREASED        2
30000 - 39999, 7001    7512  ROTOR BLADING        SEIZED, JAMMED             TEMPORARY REPAIR                1
30000 - 39999, 7001    7512  ROTOR DISCS          CRACKED                    COMPONENT RENEWED               1
30000 - 39999, 7001    7512  ROTOR DISCS          MECH. ACTION               COMPONENT RENEWED               1
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30000 - 39999  7001    7512  THROTTLE, NOZZLE VALVE          BROKEN, SHEARED              LOCATION RENEWED                 1
30000 - 39999  7001    7512  THROTTLE, NOZZLE VALVE          FAILED, DEFECTIVE            UNSPEC. REPAIR                   1
30000 - 39999  7001    7512  THRUST BRNGS. (ROTOR)           BROKEN, SHEARED              LOCATION RENEWED                 1
30000 - 39999  7001    7512  THRUST BRNGS. (ROTOR)           BURNT/SQUEEZED/HAMMERED      DEFERRED                         1
30000 - 39999  7001    7512  THRUST BRNGS. (ROTOR)           BURNT/SQUEEZED/HAMMERED      LOCATION RENEWED                 1
30000 - 39999  7001    7512  THRUST BRNGS. (ROTOR)           FAILED, DEFECTIVE            RENEWED-MODIFIED DESIGN          1
30000 - 39999  7001    7512  THRUST BRNGS. (ROTOR)           MECH. ACTION                 MACHINED, POLISHED, HONED        1
30000 - 39999  7001    7512  THRUST BRNGS. (ROTOR)           SCALED,CHOKED                LOCATION RENEWED                 1
30000 - 39999  7001    7512  THRUST COLLAR (ROTOR)           MECH. ACTION                 DEFERRED                         1
30000 - 39999  7001    7512  TUBES (CONDENSER)               CHEM. ACTION                 LOCATION RENEWED                 4
30000 - 39999  7001    7512  TUBES (CONDENSER)               CHEM. ACTION                 MISC.                            1
30000 - 39999  7001    7512  TUBES (CONDENSER)               MECH. ACTION                 LOCATION RENEWED                 3
30000 - 39999  7001    7512  TUBES (CONDENSER)               POROUS, LEAKING, HOLED       DEFERRED                         1
30000 - 39999  7001    7512  TUBES (CONDENSER)               POROUS, LEAKING, HOLED       LOCATION RENEWED                 4
30000 - 39999  7001    7512  TUBES (CONDENSER)               POROUS, LEAKING, HOLED       TEMPORARY REPAIR                 3
30000 - 39999  7001    7512  TUBES (CONDENSER)               POROUS, LEAKING, HOLED       WELDED                           1
30000 - 39999  7001    7512  TUBES (CONDENSER)               WORN, TAPERED                LOCATION RENEWED                 1
30000 - 39999  7001    7512  WATERBOX (CONDENSER)            CHEM. ACTION                 WELDED                           1
30000 - 39999  7001    7512  WATERBOX (CONDENSER)            POROUS, LEAKING, HOLED       WELDED                           1
30000 - 39999  7601    7912  BLED STEAM VALVES               CHEM. ACTION                 WELDED                           1
30000 - 39999  7601    7912  BLED STEAM VALVES               CRACKED                      WELDED                           1
30000 - 39999  7601    7912  BODY OF CASING (TURBINE)        MECH. ACTION                 WELDED                           1
30000 - 39999  7601    7912  CASING BLADING                  BROKEN, SHEARED              DEFERRED                         1
30000 - 39999  7601    7912  CASING BLADING                  BROKEN, SHEARED              FAIRED, ADJUSTED, EXPANDED       1
30000 - 39999  7601    7912  CASING BLADING                  FAILED, DEFECTIVE            MISC.                            1
30000 - 39999  7601    7912  CASING BLADING                  FLAKING,STRIPPED,TORN        MISC.                            1
30000 - 39999  7601    7912  CASING BLADING                  MECH. ACTION                 MACHINED, POLISHED, HONED        3
30000 - 39999  7601    7912  CASING BLADING                  MISALIGNED, DEFORMED         DEFERRED                         1
30000 - 39999  7601    7912  COLLAR NUT & DOWELS (ROTOR)     CRACKED                      LOCATION RENEWED                 1
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QUERY RESULTS                                                                                                              PAGE    12
  30000 - 39999  7601    7912  DIAPHRAGMS                        LOOSE, TWISTED, SLIPPED  FAIRED, ADJUSTED, EXPANDED          1
  30000 - 39999  7601    7912  DIAPHRAGMS                        MECH. ACTION             LOCATION RENEWED                    1
  30000 - 39999  7601    7912  DIAPHRAGMS                        MECH. ACTION             MACHINED, POLISHED, HONED           1
  30000 - 39999  7601    7912  DIAPHRAGMS                        MISALIGNED, DEFORMED     FAIRED, ADJUSTED, EXPANDED          1
  30000 - 39999  7601    7912  DIAPHRAGMS                        MISALIGNED, DEFORMED     MACHINED, POLISHED, HONED           1
  30000 - 39999  7601    7912  FLANGE COUPLING (ROTOR)           MECH. ACTION             LOCATION RENEWED                    1
  30000 - 39999  7601    7912  GENERAL (CONTAMINATION)(L.O. System) FAILED, DEFECTIVE     MISC.                               1
  30000 - 39999  7601    7912  GLAND BOX                         CRACKED                  LOCATION RENEWED                    1
  30000 - 39999  7601    7912  JOURNAL BRNGS. LINING (ROTOR)     BURNT/SQUEEZED/HAMMERED  LOCATION RENEWED                    5
  30000 - 39999  7601    7912  JOURNAL BRNGS. LINING (ROTOR)     BURNT/SQUEEZED/HAMMERED  MACHINED, POLISHED, HONED /         1
  30000 - 39999  7601    7912  JOURNAL BRNGS. LINING (ROTOR)     CRACKED                  LOCATION RENEWED /                  1
  30000 - 39999  7601    7912  JOURNAL BRNGS. LINING (ROTOR)     MECH. ACTION             LOCATION RENEWED                    2
  30000 - 39999  7601    7912  JOURNALS (ROTOR)                  CRACKED                  MACHINED, POLISHED, HONED /         1
  30000 - 39999  7601    7912  JOURNALS (ROTOR)                  MECH. ACTION             LOCATION RENEWED /                  1
  30000 - 39999  7601    7912  JOURNALS (ROTOR)                  MECH. ACTION             MACHINED, POLISHED, HONED /         2
  30000 - 39999  7601    7912  LABYRINTH & DUMMY PACKING         CRACKED                  LOCATION RENEWED                    1
  30000 - 39999  7601    7912  LABYRINTH & DUMMY PACKING         FAILED, DEFECTIVE        UNSPEC. REPAIR /                    1
  30000 - 39999  7601    7912  LABYRINTH & DUMMY PACKING         MECH. ACTION             MACHINED, POLISHED, HONED /         1
  30000 - 39999  7601    7912  LUB. OIL PUMP                     FAILED, DEFECTIVE        DEFERRED /                          1
  30000 - 39999  7601    7912  MANOEUVRING VALVES                CRACKED                  WELDED /                            1
  30000 - 39999  7601    7912  MANOEUVRING VALVES                FAILED, DEFECTIVE        LOCATION RENEWED                    1
  30000 - 39999  7601    7912  MANOEUVRING VALVES                SCALED,CHOKED            MISC.                               1
  30000 - 39999  7601    7912  MISC. (SPEC.) VALVES              FAILED, DEFECTIVE        UNSPEC. REPAIR /                    2
  30000 - 39999  7601    7912  MISC. (SPEC.) VALVES              LOOSE, TWISTED, SLIPPED  UNSPEC. REPAIR /                    1
  30000 - 39999  7601    7912  NOZZLE BOX                        CRACKED                  PT. RENWD-MOD. DESGN/MODIFIED       1
  30000 - 39999  7601    7912  NOZZLE BOX                        MECH. ACTION             LOCATION RENEWED /                  1
  30000 - 39999  7601    7912  NOZZLE PLATE                      BROKEN, SHEARED          LOCATION RENEWED                    1
  30000 - 39999  7601    7912  NOZZLE PLATE                      MECH. ACTION             LOCATION RENEWED                    1
  30000 - 39999  7601    7912  ROTOR BLADING                     BROKEN, SHEARED          LOCATION RENEWED /                  3
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QUERY DETAILS                                                                                     PAGE   13

30000 - 39999  7601     7912  ROTOR BLADING           BROKEN, SHEARED             MISC.                          2
30000 - 39999  7601     7912  ROTOR BLADING           BROKEN, SHEARED             TEMPORARY REPAIR               1
30000 - 39999  7601     7912  ROTOR BLADING           CRACKED                     DEFERRED                       1
30000 - 39999  7601     7912  ROTOR BLADING           CRACKED                     FAIRED, ADJUSTED, EXPANDED     1
30000 - 39999  7601     7912  ROTOR BLADING           CRACKED                     MACHINED, POLISHED, HONED      1
30000 - 39999  7601     7912  ROTOR BLADING           FAILED, DEFECTIVE           LOCATION RENEWED               1
30000 - 39999  7601     7912  ROTOR BLADING           FAILED, DEFECTIVE           MACHINED, POLISHED, HONED      1
30000 - 39999  7601     7912  ROTOR BLADING           FAILED, DEFECTIVE           MISC.                          2
30000 - 39999  7601     7912  ROTOR BLADING           FLAKING,STRIPPED,TORN       LOCATION RENEWED               1
30000 - 39999  7601     7912  ROTOR BLADING           LOOSE, TWISTED, SLIPPED     MACHINED, POLISHED, HONED      1
30000 - 39999  7601     7912  ROTOR BLADING           MECH. ACTION                LOCATION RENEWED               1
30000 - 39999  7601     7912  ROTOR BLADING           MISALIGNED, DEFORMED        FAIRED, ADJUSTED, EXPANDED     1
30000 - 39999  7601     7912  ROTOR BLADING           MISALIGNED, DEFORMED        MACHINED, POLISHED, HONED      1
30000 - 39999  7601     7912  ROTOR BLADING           MISALIGNED, DEFORMED        UNSPEC. REPAIR                 1
30000 - 39999  7601     7912  ROTOR DISCS             MECH. ACTION                MACHINED, POLISHED, HONED      1
30000 - 39999  7601     7912  ROTOR DISCS             MECH. ACTION                UNSPEC. REPAIR                 1
30000 - 39999  7601     7912  THRUST BRNGS. (ROTOR)   BROKEN, SHEARED             LOCATION RENEWED               1
30000 - 39999  7601     7912  THRUST BRNGS. (ROTOR)   BURNT/SQUEEZED/HAMMERED     LOCATION RENEWED               1
30000 - 39999  7601     7912  THRUST BRNGS. (ROTOR)   MECH. ACTION                DEFERRED                       1
30000 - 39999  7601     7912  THRUST BRNGS. (ROTOR)   MECH. ACTION                LOCATION RENEWED               1
30000 - 39999  7601     7912  TUBEPLATE (CONDENSER)   MECH. ACTION                TEMPORARY REPAIR               2
30000 - 39999  7601     7912  TUBES (CONDENSER)       CHEM. ACTION                LOCATION RENEWED               2
30000 - 39999  7601     7912  TUBES (CONDENSER)       FAILED, DEFECTIVE           LOCATION RENEWED               1
30000 - 39999  7601     7912  TUBES (CONDENSER)       MECH. ACTION                CHEM. TRTMENT, DEGREASED       1
30000 - 39999  7601     7912  TUBES (CONDENSER)       MECH. ACTION                LOCATION RENEWED               2
30000 - 39999  7601     7912  TUBES (CONDENSER)       MECH. ACTION                TEMPORARY REPAIR               1
30000 - 39999  7601     7912  TUBES (CONDENSER)       POROUS, LEAKING, HOLED      TEMPORARY REPAIR               1
30000 - 39999  7601     7912  TUBES (CONDENSER)       SCALED,CHOKED               LOCATION RENEWED               1
30000 - 39999  7601     7912  TUBES (CONDENSER)       SCALED,CHOKED               MISC.                          2
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[illegible]                                                                                                   PAGE    5
 20000 - 29999  7001   7512  TUBES (CONDENSER)                  FAILED, DEFECTIVE          LOCATION RENEWED /                1
 20000 - 29999  7001   7512  TUBES (CONDENSER)                  MECH. ACTION               DEFERRED /                        1
 20000 - 29999  7001   7512  TUBES (CONDENSER)                  MISALIGNED, DEFORMED       LOCATION RENEWED ✓                1
 20000 - 29999  7001   7512  TUBES (CONDENSER)                  POROUS, LEAKING, HOLED     LOCATION RENEWED                  3
 20000 - 29999  7001   7512  TUBES (CONDENSER)                  POROUS, LEAKING, HOLED     TEMPORARY REPAIR ✓                2
 20000 - 29999  7001   7512  TUBES (CONDENSER)                  SCALED,CHOKED              LOCATION RENEWED                  1
 20000 - 29999  7001   7512  UNSPEC (CONDENSER)                 CHEM. ACTION               MISC. /                           2
 20000 - 29999  7001   7512  UNSPEC (CONDENSER)                 SCALED,CHOKED              MISC. /                           2
 20000 - 29999  7001   7512  WATERBOX (CONDENSER)               CHEM. ACTION               METL SPRAYED/PLAST STL/METL /     2
 20000 - 29999  7001   7512  WATERBOX (CONDENSER)               CHEM. ACTION               REINFORCED/STRAPPED /             1
 20000 - 29999  7601   7912  BODY OF CASING (TURBINE)           CRACKED                    DEFERRED /                        1
 20000 - 29999  7601   7912  CASING BLADING                     BROKEN, SHEARED            PT. RENWD-MOD. DESGN/MODIFIED /   1
 20000 - 29999  7601   7912  CASING BLADING                     CRACKED                    DEFERRED /                        1
 20000 - 29999  7601   7912  CASING BLADING                     FAILED, DEFECTIVE          MISC. ✓                           1
 20000 - 29999  7601   7912  CASING BLADING                     FAILED, DEFECTIVE          UNSPEC. REPAIR /                  1
 20000 - 29999  7601   7912  CASING BLADING                     LOOSE, TWISTED, SLIPPED    COMPONENT RENEWED ✓               1
 20000 - 29999  7601   7912  CASING BLADING                     MECH. ACTION               LOCATION RENEWED /                1
 20000 - 29999  7601   7912  CASING BLADING                     MECH. ACTION               TEMPORARY REPAIR ✓                1
 20000 - 29999  7601   7912  CASING BLADING                     MISALIGNED, DEFORMED       COMPONENT RENEWED X               1
 20000 - 29999  7601   7912  CASING BLADING                     MISALIGNED, DEFORMED       MACHINED, POLISHED, HONED /       1
 20000 - 29999  7601   7912  CASING BOLTS (TURBINE)             BROKEN, SHEARED            LOCATION RENEWED ✓                1
 20000 - 29999  7601   7912  CASING FLANGES (TURBINE)           CRACKED                    UNSPEC. REPAIR /                  1
 20000 - 29999  7601   7912  DIAPHRAGMS                         CRACKED                    MACHINED, POLISHED, HONED X       1
 20000 - 29999  7601   7912  DIAPHRAGMS                         FAILED, DEFECTIVE          LOCATION RENEWED ✓                1
 20000 - 29999  7601   7912  DIAPHRAGMS                         MECH. ACTION               MISC. /                           1
 20000 - 29999  7601   7912  DIAPHRAGMS                         MECH. ACTION               TEMPORARY REPAIR ✓                1
 20000 - 29999  7601   7912  GENERAL (CONTAMINATION)(L.O.I.S.etc)FAILED, DEFECTIVE         LOCATION RENEWED                  1
 20000 - 29999  7601   7912  GLAND BOX                          MECH. ACTION               LOCATION RENEWED                  1
 20000 - 29999  7601   7912  GLAND BOX                          POROUS, LEAKING, HOLED     LOCATION RENEWED                  1
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QUERY HOLDFL3                                                                                              PAGE    4

20000 - 29999  7001    7512  ROTOR  BLADING             MECH. ACTION              FAIRED. ADJUSTED. EXPANDED          1
20000 - 29999  7001    7512  ROTOR  BLADING             MECH. ACTION              MACHINED. POLISHED. HONED           2
20000 - 29999  7001    7512  ROTOR  BLADING             MECH. ACTION              MISC.                               2
20000 - 29999  7001    7512  ROTOR  BLADING             MECH. ACTION              TEMPORARY REPAIR                    1
20000 - 29999  7001    7512  ROTOR  BLADING             MISALIGNED. DEFORMED      COMPONENT RENEWED                   1
20000 - 29999  7001    7512  ROTOR  BLADING             MISALIGNED. DEFORMED      DEFERRED                            1
20000 - 29999  7001    7512  ROTOR  BLADING             MISALIGNED. DEFORMED      MACHINED. POLISHED. HONED           2
20000 - 29999  7001    7512  ROTOR  BLADING             MISALIGNED. DEFORMED      MISC.                               2
20000 - 29999  7001    7512  ROTOR  BLADING             MISALIGNED. DEFORMED      PT. RENWD-MOD. DESGN/MODIFIED       1
20000 - 29999  7001    7512  ROTOR  BLADING             MISALIGNED. DEFORMED      TEMPORARY REPAIR                    1
20000 - 29999  7001    7512  ROTOR  BLADING             MISSING. LOST             LOCATION RENEWED                    1
20000 - 29999  7001    7512  ROTOR  BLADING             MISSING. LOST             MISC.                               1
20000 - 29999  7001    7512  ROTOR  BLADING             SCALED.CHOKED             MACHINED. POLISHED. HONED           1
20000 - 29999  7001    7512  ROTOR  BLADING             VIBRATION                 MACHINED. POLISHED. HONED           1
20000 - 29999  7001    7512  SHELL  (CONDENSER)         CHEM. ACTION              LOCATION RENEWED                    2
20000 - 29999  7001    7512  SHELL  (CONDENSER)         POROUS. LEAKING. HOLED    METL SPRAYED/PLAST STL/METL         1
20000 - 29999  7001    7512  THRUST BRNGS.(ROTOR)       BURNT/SQUEEZED/HAMMERED   DEFERRED                            1
20000 - 29999  7001    7512  THRUST BRNGS. (ROTOR)      BURNT/SQUEEZED/HAMMERED   LOCATION RENEWED                    1
20000 - 29999  7001    7512  THRUST BRNGS. (ROTOR)      BURNT/SQUEEZED/HAMMERED   MACHINED. POLISHED. HONED           2
20000 - 29999  7001    7512  THRUST BRNGS. (ROTOR)      CRACKED                   LOCATION RENEWED                    1
20000 - 29999  7001    7512  THRUST BRNGS. (ROTOR)      FAILED. DEFECTIVE         DEFERRED                            2
20000 - 29999  7001    7512  THRUST BRNGS. (ROTOR)      MECH. ACTION              LOCATION RENEWED                    1
20000 - 29999  7001    7512  THRUST COLLAR (ROTOR)      BROKEN. SHEARED           TEMPORARY REPAIR                    1
20000 - 29999  7001    7512  THRUST COLLAR (ROTOR)      LOOSE. TWISTED. SLIPPED   PT. RENWD-MOD. DESGN/MODIFIED       1
20000 - 29999  7001    7512  THRUST COLLAR (ROTOR)      MECH. ACTION              LOCATION RENEWED                    1
20000 - 29999  7001    7512  THRUST COLLAR (ROTOR)      MECH. ACTION              MACHINED. POLISHED. HONED           3
20000 - 29999  7001    7512  TUBES (CONDENSER)          CHEM. ACTION              LOCATION RENEWED                    1
20000 - 29999  7001    7512  TUBES (CONDENSER)          CHEM. ACTION              MISC.                               1
20000 - 29999  7001    7512  TUBES (CONDENSER)          CRACKED                   LOCATION RENEWED                    1
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QUERY HOLDERS
                                                                                                          PAGE    3
20000 - 29999  7001   7512  NOZZLE BOX        MECH. ACTION               COMPONENT RENEWED                    1
20000 - 29999  7001   7512  NOZZLE PLATE      CHEM. ACTION               LOCATION RENEWED ✓                   1
20000 - 29999  7001   7512  NOZZLE PLATE      CRACKED                    MACHINED, POLISHED, HONED ✓          2
20000 - 29999  7001   7512  NOZZLE PLATE      FAILED, DEFECTIVE          MACHINED, POLISHED, HONED ✓          1
20000 - 29999  7001   7512  NOZZLE PLATE      MISALIGNED, DEFORMED       LOCATION RENEWED ✓                   1
20000 - 29999  7001   7512  ROTOR BLADING     BROKEN, SHEARED            COMPONENT RENEWED ✓                  1
20000 - 29999  7001   7512  ROTOR BLADING     BROKEN, SHEARED            DEFERRED X                           1
20000 - 29999  7001   7512  ROTOR BLADING     BROKEN, SHEARED            FAIRED, ADJUSTED, EXPANDED           1
20000 - 29999  7001   7512  ROTOR BLADING     BROKEN, SHEARED            LOCATION RENEWED                     4
20000 - 29999  7001   7512  ROTOR BLADING     BROKEN, SHEARED            MACHINED, POLISHED, HONED            1
20000 - 29999  7001   7512  ROTOR BLADING     BROKEN, SHEARED            TEMPORARY REPAIR ✓                   1
20000 - 29999  7001   7512  ROTOR BLADING     CHEM. ACTION               MACHINED, POLISHED, HONED ✓          1
20000 - 29999  7001   7512  ROTOR BLADING     CHEM. ACTION               MISC. ✓                              1
20000 - 29999  7001   7512  ROTOR BLADING     CRACKED                    COMPONENT RENEWED ✓                  1
20000 - 29999  7001   7512  ROTOR BLADING     CRACKED                    FAIRED, ADJUSTED, EXPANDED           1
20000 - 29999  7001   7512  ROTOR BLADING     CRACKED                    LOCATION RENEWED ✓                   1
20000 - 29999  7001   7512  ROTOR BLADING     CRACKED                    MACHINED, POLISHED, HONED ✓          4
20000 - 29999  7001   7512  ROTOR BLADING     CRACKED                    MISC. ✓                              2
20000 - 29999  7001   7512  ROTOR BLADING     CRACKED                    TEMPORARY REPAIR ✓                   1
20000 - 29999  7001   7512  ROTOR BLADING     CRACKED                    UNSPEC. REPAIR ✓                     1
20000 - 29999  7001   7512  ROTOR BLADING     FAILED, DEFECTIVE          DEFERRED X                           1
20000 - 29999  7001   7512  ROTOR BLADING     FAILED, DEFECTIVE          LOCATION RENEWED ✓                   2
20000 - 29999  7001   7512  ROTOR BLADING     FAILED, DEFECTIVE          MACHINED, POLISHED, HONED            1
20000 - 29999  7001   7512  ROTOR BLADING     FAILED, DEFECTIVE          PT. RENWD-MOD. DESGN/MODIFIED ✓      1
20000 - 29999  7001   7512  ROTOR BLADING     FAILED, DEFECTIVE          UNSPEC. REPAIR ✓                     1
20000 - 29999  7001   7512  ROTOR BLADING     FLAKING,STRIPPED,TORN      FAIRED, ADJUSTED, EXPANDED ✓         1
20000 - 29999  7001   7512  ROTOR BLADING     FLAKING,STRIPPED,TORN      LOCATION RENEWED ✓                   1
20000 - 29999  7001   7512  ROTOR BLADING     FLAKING,STRIPPED,TORN      MACHINED, POLISHED, HONED ✓          2
20000 - 29999  7001   7512  ROTOR BLADING     LOOSE, TWISTED, SLIPPED    LOCATION RENEWED ✓                   1
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QUERY HOLDFL3                                                                                            PAGE      2
20000 - 29999  7001     7512  JOURNAL BRNGS. LINING (ROTOR)   BURNT/SQUEEZED/HAMMERED  LOCATION RENEWED                   3
20000 - 29999  7001     7512  JOURNAL BRNGS. LINING (ROTOR)   BURNT/SQUEEZED/HAMMERED  MACHINED, POLISHED, HONED          5
20000 - 29999  7001     7512  JOURNAL BRNGS. LINING (ROTOR)   CRACKED                  COMPONENT RENEWED                  1
20000 - 29999  7001     7512  JOURNAL BRNGS. LINING (ROTOR)   CRACKED                  LOCATION RENEWED                   3
20000 - 29999  7001     7512  JOURNAL BRNGS. LINING (ROTOR)   DISINTEGRATED            TEMPORARY REPAIR                   1
20000 - 29999  7001     7512  JOURNAL BRNGS. LINING (ROTOR)   FAILED, DEFECTIVE        LOCATION RENEWED                   1
20000 - 29999  7001     7512  JOURNAL BRNGS. LINING (ROTOR)   LOOSE, TWISTED, SLIPPED  COMPONENT RENEWED                  1
20000 - 29999  7001     7512  JOURNAL BRNGS. LINING (ROTOR)   LOOSE, TWISTED, SLIPPED  LOCATION RENEWED                   2
20000 - 29999  7001     7512  JOURNAL BRNGS. LINING (ROTOR)   MECH. ACTION             MISC.                              1
20000 - 29999  7001     7512  JOURNALS (ROTOR)                BURNT/SQUEEZED/HAMMERED  LOCATION RENEWED                   1
20000 - 29999  7001     7512  JOURNALS (ROTOR)                MECH. ACTION             LOCATION RENEWED                   3
20000 - 29999  7001     7512  JOURNALS (ROTOR)                MECH. ACTION             MACHINED, POLISHED, HONED          6
20000 - 29999  7001     7512  JOURNALS (ROTOR)                MECH. ACTION             MISC.                              1
20000 - 29999  7001     7512  JOURNALS (ROTOR)                WORN, TAPERED            MACHINED, POLISHED, HONED          1
20000 - 29999  7001     7512  LABYRINTH & DUMMY PACKING       FAILED, DEFECTIVE        LOCATION RENEWED                   1
20000 - 29999  7001     7512  LABYRINTH & DUMMY PACKING       SCALED,CHOKED            MISC.                              2
20000 - 29999  7001     7512  LABYRINTH & DUMMY PACKING       SEIZED, JAMMED           LOCATION RENEWED                   1
20000 - 29999  7001     7512  LABYRINTH & DUMMY PACKING       SEIZED, JAMMED           MISC.                              2
20000 - 29999  7001     7512  MANOEUVRING VALVES              CRACKED                  LOCATION RENEWED                   2
20000 - 29999  7001     7512  MANOEUVRING VALVES              CRACKED                  WELDED                             4
20000 - 29999  7001     7512  MANOEUVRING VALVES              FAILED, DEFECTIVE        LOCATION RENEWED                   1
20000 - 29999  7001     7512  MANOEUVRING VALVES              MISALIGNED, DEFORMED     LOCATION RENEWED                   1
20000 - 29999  7001     7512  MISC. (SPEC.) VALVES            BROKEN, SHEARED          TEMPORARY REPAIR                   1
20000 - 29999  7001     7512  MISC. (SPEC.) VALVES            BURNT/SQUEEZED/HAMMERED  LOCATION RENEWED                   1
20000 - 29999  7001     7512  MISC. (SPEC.) VALVES            FAILED, DEFECTIVE        NONE                               1
20000 - 29999  7001     7512  MISC. (SPEC.) VALVES            MISALIGNED, DEFORMED     LOCATION RENEWED                   1
20000 - 29999  7001     7512  MISC. (SPEC.) VALVES            POROUS, LEAKING, HOLED   WELDED                             1
20000 - 29999  7001     7512  MISC. (SPEC.) VALVES            SEIZED, JAMMED           LOCATION RENEWED                   1
20000 - 29999  7001     7512  NOZZLE BOX                      CHEM. ACTION             LOCATION RENEWED                   1
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QUERY NUMBER 1

STEAM TURBINES AND CONDENSERS WITH AN SHP OF 20000 AND OVER FITTED TO SHIPS BUILT TO LR CLASS BETWEEN JANUARY 1960 AND DECEMBER 1979

SHP RANGE	TIME INTERVAL		DEFECTIVE LOCATION	DEFECT MODE	REPAIR	COUNT
20000 - 29999	7001	7512	BODY OF CASING (TURBINE)	CRACKED	DEFERRED	1
20000 - 29999	7001	7512	BODY OF CASING (TURBINE)	CRACKED	METALOCKED	1
20000 - 29999	7001	7512	CASING BLADING	BROKEN, SHEARED	LOCATION RENEWED	1
20000 - 29999	7001	7512	CASING BLADING	CHEM. ACTION	LOCATION RENEWED	1
20000 - 29999	7001	7512	CASING BLADING	CHEM. ACTION	MACHINED, POLISHED, HONED	1
20000 - 29999	7001	7512	CASING BLADING	FAILED, DEFECTIVE	LOCATION RENEWED	1
20000 - 29999	7001	7512	CASING BLADING	FAILED, DEFECTIVE	UNSPEC. REPAIR	1
20000 - 29999	7001	7512	CASING BLADING	FLAKING, STRIPPED, TORN	LOCATION RENEWED	1
20000 - 29999	7001	7512	CASING BLADING	MISALIGNED, DEFORMED	COMPONENT RENEWED	1
20000 - 29999	7001	7512	CASING BLADING	MISALIGNED, DEFORMED	MACHINED, POLISHED, HONED	1
20000 - 29999	7001	7512	CASING BLADING	MISALIGNED, DEFORMED	TEMPORARY REPAIR	1
20000 - 29999	7001	7512	CASING BLADING	MISSING, LOST	UNSPEC. REPAIR	1
20000 - 29999	7001	7512	CASING BOLTS (TURBINE)	BROKEN, SHEARED	LOCATION RENEWED	1
20000 - 29999	7001	7512	CASING BOLTS (TURBINE)	CRACKED	LOCATION RENEWED	2
20000 - 29999	7001	7512	DIAPHRAGMS	CRACKED	MACHINED, POLISHED, HONED	1
20000 - 29999	7001	7512	DIAPHRAGMS	MECH. ACTION	LOCATION RENEWED	1
20000 - 29999	7001	7512	DIAPHRAGMS	MECH. ACTION	MACHINED, POLISHED, HONED	1
20000 - 29999	7001	7512	DOOR (CONDENSER)	CHEM. ACTION	REINFORCED/STRAPPED	1
20000 - 29999	7001	7512	DOOR (CONDENSER)	CRACKED	COMPONENT RENEWED	1
20000 - 29999	7001	7512	FLANGE COUPLING (ROTOR)	DISINTEGRATED	TEMPORARY REPAIR	1
20000 - 29999	7001	7512	JOURNAL BRNGS. HOUSING (ROTOR)	BURNT/SQUEEZED/HAMMERED	LOCATION RENEWED	1
20000 - 29999	7001	7512	JOURNAL BRNGS. HOUSING (ROTOR)	MECH. ACTION	MACHINED, POLISHED, HONED	2
20000 - 29999	7001	7512	JOURNAL BRNGS. HOUSING (ROTOR)	MISALIGNED, DEFORMED	MACHINED, POLISHED, HONED	1
20000 - 29999	7001	7512	JOURNAL BRNGS. LINING (ROTOR)	BROKEN, SHEARED	LOCATION RENEWED	1
20000 - 29999	7001	7512	JOURNAL BRNGS. LINING (ROTOR)	BURNT/SQUEEZED/HAMMERED	COMPONENT RENEWED	1

```
QUERY HOLDELS                                                                                   PAGE     1

STEAM TURBINES AND CONDENSERS WITH AN SHP OF 20000 AND OVER FITTED TO SHIPS BUILT TO LR CLASS BETWEEN JANUARY 1960 AND DECEMBER 1979

        SHP RANGE        TIME INTERVAL     NO. OF   NO. AT    MONTHS     INCIDENCE

                                           DEFECTS   RISK     SERVICE      RATE


        20000 - 29999     7001     7512      96      125       7756         1.24

        20000 - 29999     7601     7912      50      111       3460         1.45

        30000 - 39999     7001     7512      65      142       5186         1.25

        30000 - 39999     7601     7912      58      196       7867         0.74

        40000 - 49999     7001     7512      11       34       1297         0.85

        40000 - 49999     7601     7912      15       46       1651         0.91

        50000 PLUS        7001     7512       2        2        144         1.39

        50000 PLUS        7601     7912       2        2         96         2.08



     N.B.     1) A DEFECT IS COUNTED ONCE AS AN INCIDENT REPORTED AT A POINT IN TIME

                 REGARDLESS OF THE EXTENT OR SEVERITY

              2) SERVICE TIME IS CALENDER TIME EXCLUDING LAY-UP TIME

              3) INCIDENCE RATE IS THE RATIO OF NUMBER OF DEFECTS TO THE AGGREGATE

                 SERVICE TIMES 100
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QUERY RESULTS
                                                                                                  PAGE    14
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      CHEM. ACTION               DEFERRED                          1
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      CHEM. ACTION               LOCATION RENEWED                  1
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      CHEM. ACTION               METL SPRAYED/PLAST STL/METL ✓     1
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      CHEM. ACTION               REINFORCED/STRAPPED               2
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      CHEM. ACTION               TEMPORARY REPAIR ✓                2
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      CHEM. ACTION               UNSPEC. REPAIR                    2
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      CRACKED                    DEFERRED                          1
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      FAILED, DEFECTIVE          MISC.                             1
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      MECH. ACTION               TEMPORARY REPAIR ✓                1
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      POROUS, LEAKING, HOLED     REINFORCED/STRAPPED               1
30000 - 39999 7601   7912 WATERBOX (CONDENSER)      POROUS, LEAKING, HOLED     TEMPORARY REPAIR                  3
40000 - 49999 7001   7512 CASING BLADING            BROKEN, SHEARED            TEMPORARY REPAIR ✓                1
40000 - 49999 7001   7512 CASING BLADING            FAILED, DEFECTIVE          MACHINED, POLISHED, HONED ✓       1
40000 - 49999 7001   7512 CASING BLADING            MECH. ACTION               TEMPORARY REPAIR ✓                1
40000 - 49999 7001   7512 GLAND BOX                 SCALED,CHOKED              CHEM. TRTMENT, DEGREASED          1
40000 - 49999 7001   7512 MISC. (SPEC.) (ROTOR)     MISALIGNED, DEFORMED       FAIRED, ADJUSTED, EXPANDED ✓      1
40000 - 49999 7001   7512 MISC. (SPEC.) (ROTOR)     VIBRATION                  UNSPEC. REPAIR ✓                  1
40000 - 49999 7001   7512 NOZZLE BOX                CRACKED                    WELDED ✓                          1
40000 - 49999 7001   7512 NOZZLE PLATE              BROKEN, SHEARED            LOCATION RENEWED                  1
40000 - 49999 7001   7512 NOZZLE PLATE              CRACKED                    LOCATION RENEWED                  1
40000 - 49999 7001   7512 NOZZLE PLATE              FAILED, DEFECTIVE          MACHINED, POLISHED, HONED ✓       1
40000 - 49999 7001   7512 NOZZLE PLATE              FLAKING,STRIPPED,TORN      LOCATION RENEWED                  1
40000 - 49999 7001   7512 NOZZLE PLATE              POROUS, LEAKING, HOLED     MISC. ✓                           2
40000 - 49999 7001   7512 ROTOR BLADING             BROKEN, SHEARED            LOCATION RENEWED                  1
40000 - 49999 7001   7512 ROTOR BLADING             BROKEN, SHEARED            TEMPORARY REPAIR                  1
40000 - 49999 7001   7512 ROTOR BLADING             FAILED, DEFECTIVE          LOCATION RENEWED                  1
40000 - 49999 7001   7512 ROTOR BLADING             FAILED, DEFECTIVE          MACHINED, POLISHED, HONED ✓       1
40000 - 49999 7001   7512 ROTOR BLADING             LOOSE, TWISTED, SLIPPED    FAIRED, ADJUSTED, EXPANDED        1
40000 - 49999 7001   7512 ROTOR BLADING             LOOSE, TWISTED, SLIPPED    LOCATION RENEWED ✓                1
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QUERY RESULTS
40000 - 49999  7601      7912  CASING BLADING                    FAILED, DEFECTIVE          MISC.                              1
40000 - 49999  7601      7912  GLAND BOX                         FAILED, DEFECTIVE          MACHINED, POLISHED, HONED          1
40000 - 49999  7601      7912  JOURNAL BRNGS. LINING (ROTOR)     FAILED, DEFECTIVE          LOCATION RENEWED                   1
40000 - 49999  7601      7912  LABYRINTH & DUMMY PACKING         MECH. ACTION               MACHINED, POLISHED, HONED          1
40000 - 49999  7601      7912  NOZZLE PLATE                      FAILED, DEFECTIVE          PT. RENWD-MOD. DESGN/MODIFIED      8
40000 - 49999  7601      7912  ROTOR BLADING                     BROKEN, SHEARED            COMPONENT RENEWED                  1
40000 - 49999  7601      7912  ROTOR BLADING                     BROKEN, SHEARED            UNSPEC. REPAIR                     1
40000 - 49999  7601      7912  ROTOR BLADING                     FAILED, DEFECTIVE          COMPONENT RENEWED                  4
40000 - 49999  7601      7912  ROTOR BLADING                     FAILED, DEFECTIVE          LOCATION RENEWED                   4
40000 - 49999  7601      7912  ROTOR BLADING                     FAILED, DEFECTIVE          MISC.                              1
40000 - 49999  7601      7912  THRUST BRNGS. (ROTOR)             MISALIGNED, DEFORMED       LOCATION RENEWED                   1
40000 - 49999  7601      7912  TUBES (CONDENSER)                 CHEM. ACTION               LOCATION RENEWED                   1
40000 - 49999  7601      7912  TUBES (CONDENSER)                 FAILED, DEFECTIVE          TEMPORARY REPAIR                   1
40000 - 49999  7601      7912  TUBES (CONDENSER)                 MECH. ACTION               LOCATION RENEWED                   1
40000 - 49999  7601      7912  TUBES (CONDENSER)                 POROUS, LEAKING, HOLED     LOCATION RENEWED                   1
40000 - 49999  7601      7912  TUBES (CONDENSER)                 POROUS, LEAKING, HOLED     TEMPORARY REPAIR                   1
40000 - 49999  7601      7912  TUBES (CONDENSER)                 SCALED,CHOKED              TEMPORARY REPAIR                   1
40000 - 49999  7601      7912  UNSPEC (CONDENSER)                MISALIGNED, DEFORMED       COMPONENT RENEWED                  1
50000 PLUS     7001      7512  CASING BLADING                    BROKEN, SHEARED            COMPONENT RENEWED                  1
50000 PLUS     7001      7512  JOURNAL BRNGS. LINING (ROTOR)     FAILED, DEFECTIVE          COMPONENT RENEWED                  1
50000 PLUS     7001      7512  ROTOR BLADING                     CRACKED                    LOCATION RENEWED                   1
50000 PLUS     7001      7512  ROTOR BLADING                     FLAKING,STRIPPED,TORN      COMPONENT RENEWED                  1
50000 PLUS     7601      7912  CASING BLADING                    CRACKED                    MACHINED, POLISHED, HONED          1
50000 PLUS     7601      7912  CASING BLADING                    LOOSE, TWISTED, SLIPPED    MACHINED, POLISHED, HONED          1
50000 PLUS     7601      7912  ROTOR BLADING                     BROKEN, SHEARED            MISC.                              1
50000 PLUS     7601      7912  ROTOR BLADING                     MISALIGNED, DEFORMED       MACHINED, POLISHED, HONED          1
```

APPENDIX 2

Materials and Correspondence Received During the Course of the Study

CATALOGUE OF MATERIALS CONTAINED IN APPENDIX 2

1. Combined Cycle Propulsion System, Foster Wheeler Limited, A. Lozsef, Letter to J. Wainwright, February 28, 1980.
2. Conversation with H. Douglas Marron (Project Engineer, U.S. Navy RACER Program) J. Wainwright, Inter-Office Memo to file, March 5, 1980.
3. Crew Structure Combined Cycle Propulsion, J. Wainwright, Inter-Office Memo to file, March 6, 1980.
4. Combined Cycle Propulsion System, Icebreaking LNG Vessels, Sunrod ab, J. Wester, Letter to J. Wainwright, March 11, 1980.
5. Thermal Cycling of Steam Turbines, G.E.C. Turbine Generators Limited, D. G. Nicholas, Letter to J. Wainwright, March 19, 1980.
6. Rolls Royce Combined Cycle Experience, B. Tedford, Letter to J. Wainwright, April 15, 1980.
7. Letter to J. Wainwright, Solar Turbines International, G. A. Clough, September 3, 1980.
8. Waste Heat Boilers for Combined Cycle Propulsion, Minutes of Meeting to file, September 18, 1980.
9. Combined Cycle Propulsion, J. Wainwright, Inter-Office Memo to file, October 17, 1980.
10. Energy Conservation, Solar Presentation Slides, October 13, 1980.
11. Rankine Cycle Energy Recovery, U.S. Navy Program Applied to LM 2500 Engine, Garrett, October 9, 1980.
12. Russian COGAS in the Thames, Shipbuilding and Marine Engineering International, June 1980.
13. Marine Power Plant for Energy Savings, COGAS: Marine Technology, R. P. Giblon and I. H. Rolin, July 1979.
14. A Dynamic Analysis of a COGAS Propulsion Plant, SNAME, Chesapeake Section, J. W. Abbott, J. G. McIntyre and C. J. Rubis, December 7, 1976.
15. An advanced Combined Cycle Marine Power Plant for an Arctic LNG Tanker, SNAME, G. A. Clough, April 1980.

16. Stag Combined Cycle Operating Experience, General Electric, J. G. Weigel.

17. A Proposed Combined Cycle Marine Power Plant, SNAME, N. L. McIntyre, January 1979.

18. Design Aspects of an Advanced 14,000 HP. Marine Combined Cycle Plant, Solar Turbines International, March 1979.

19. Design Considerations for Heat Recovery System for DD-963 Class Ship, J. L. Boyen and Y. Katz, December 1976.

20. DD-963 Waste Heat Recovery System Experience, Rolls Royce, B. Tedford, Letter to J. Wainwright, October 31, 1979.

FOSTER WHEELER LIMITED

DESIGNERS – MANUFACTURERS – CONSTRUCTORS

P.O. BOX 3007 – ST.CATHARINES, ONTARIO L2R 7B7

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CABLE - REWOP

MONTREAL
TORONTO
CALGARY

February 28, 1980

Melville Shipping Ltd.,
1253 McGill College, Suite 301,
Montreal, Quebec,
H3B 2Y4

Attention: Mr. J. Wainwright

Reference: Combined Cycle Propulsion System
For: Arctic LNG Carrier - Ice Breaker
Foster Wheeler File No.478-149

Dear Mr. Wainwright:

Thank you for the opportunity offered to us on February 12 to present to you Foster Wheeler's experience and interest in the field of Combined Cycle Steam Generators.

Further to our discussions on that occasion, we are pleased to submit to you answers and comments to the questions listed in your letter of January 23, 1980.

1. A diagram reflecting variation of steam output, stack gas temperature, heat transfer surface and boiler cost versus steam pressure for unfired Waste Heat Boilers generating 770°F. superheated steam from 850°F. turbine exhaust gas with 40° pinch point at boiler bank outlet, is attached.

   It shows that a low steam generation pressure is to be preferred as this is granting a low stack temperature (high cycle efficiency, i.e. important fuel savings) for not very significant premium on the purchase price of the boiler.

   We consider the 40°F. pinch point suggested by you as reasonable. A smaller pinch point would increase the cost of the boiler and the gas side pressure loss.

   As a remark to your calculation sheets, we mention that:

   a) Gas turbine exhaust gas specific heats are higher than .248. As an average over the 800 to 400°F. range, is around .264.

....2

b) Water temperature at economizer outlet shall be some 50°F. below saturation to avoid steaming in the economizer at lower loads. This will lead to a higher stack temperature, i.e. 420°F. roughly.

2. We think the best arrangement in your case is one boiler for each gas turbine. Coupling two gas turbines to one boiler would require significant extra space to accommodate the duct work and also expensive and not fully reliable dampers.

3. As turbine exhaust gas is only 850°F. and there are six gas turbines, we think the best option would be without by-pass stack and damper. A small premium will be paid on boiler tubing to allow dry operation without carburization (C-1/2 Mo tubes instead of C.S. tubes). This will cost much less than exhaust stack and dampers.

4. The boiler designed to operate on waste heat only during normal cruising, can be operated with auxiliary firing during ramming cycles. The approximate performances when exhaust gas temperature is increased to 1200°F. by natural gas firing with a duct burner are:

Steam generation	100,000 PPH
Steam pressure	340 PSIG
Steam temperature	750°F.
Stack temperature	440°F.
Fuel Heat input (in LHV)	56,700,000 BTU/HR.
Extra steam output	45,000 LB/HR.
Steam output in BTU/HR. 45000 (1391-208) =	53,235,000 BTU/HR.
Firing efficiency (in LHV)	93.88%

The 1200°F. firing temperature mentioned above is considered by us to be a practical design temperature with a maximum allowable to be 1400°F. Up to these temperatures, we can build combustion chambers without internal insulation.

Enclosed are sketches showing boiler shape and sizes for both unfired and fired boilers. As you will see, only extra space required for the fired boiler is in height. An extra 15 feet in height is required to allow for the combustion chamber.

....3

An extra 1.5 to 2 inches W. Col. gas side pressure loss is to be expected if boiler designed with firing capability.

There is plenty of experience with auxiliary fired G.T. Waste Heat Boilers. Foster Wheeler reference list attached to this letter shows the majority of units supplied by our company are auxiliary fired.

Firing equipment for natural gas firing is common and reliable.

5. The boilers need only a feedwater flow control and if fired boiler selected, a steam temperature control loop is also needed. If waste heat boilers with no auxiliary firing are considered, steam turbine operation cannot be interlocked with gas turbine operation as amount of steam generated is a result of gas turbine exhaust flow and temperature. In this case, the steam turbine has to take all the steam generated by the boiler(s) but cannot initiate an increase of the steam supply. Thus, only the gas turbine has to be controled to accommodate load changes.

   With fired Waste Heat Boilers, a load change can act simultaneously both on the gas turbine and steam turbine. A change in load on the steam turbine will act on the boiler burner input control. This is valid inside a steam generation rate where burners are in operation (say 60 to 100% of full steam generation rate).

6. Cold start-up of the first boiler is lengthy, it will take 1-1/2 to 2 hours. The remaining boilers can be brought up to almost operating temperature with steam generated by the first started boiler before gas turbines are started up. To bring up a boiler, previously warmed up, to full steaming capacity, will take 10 to 20 minutes.

   When cruising with some of power groups shut down, the waste heat boilers can be kept on stand-by (at normal operating pressure and temperature level) as described above and can be on full power in 10-20 minutes.

.....4

7. The boiler's response to the fast changing load conditions cannot be predicted at this stage. In contract design stage, a computerized system response analysis has to be performed to assess the actual changes in pressure temperature and water level. Foster Wheeler does have the capability to perform this analysis and we are confident, that with approprate design and control system features, the boilers will be able to co-operate positively in the overall power system.

8. Land based combined cycle Waste Heat Boilers are very reliable. No major breakdowns were experienced and continuous operation for at least one year is common. Some investigation is needed to select a proper mechanical design for this unusual application.

   We feel confident that by combining our land based waste heat boiler experience with our marine boiler experience, a suitable and reliable design will result.

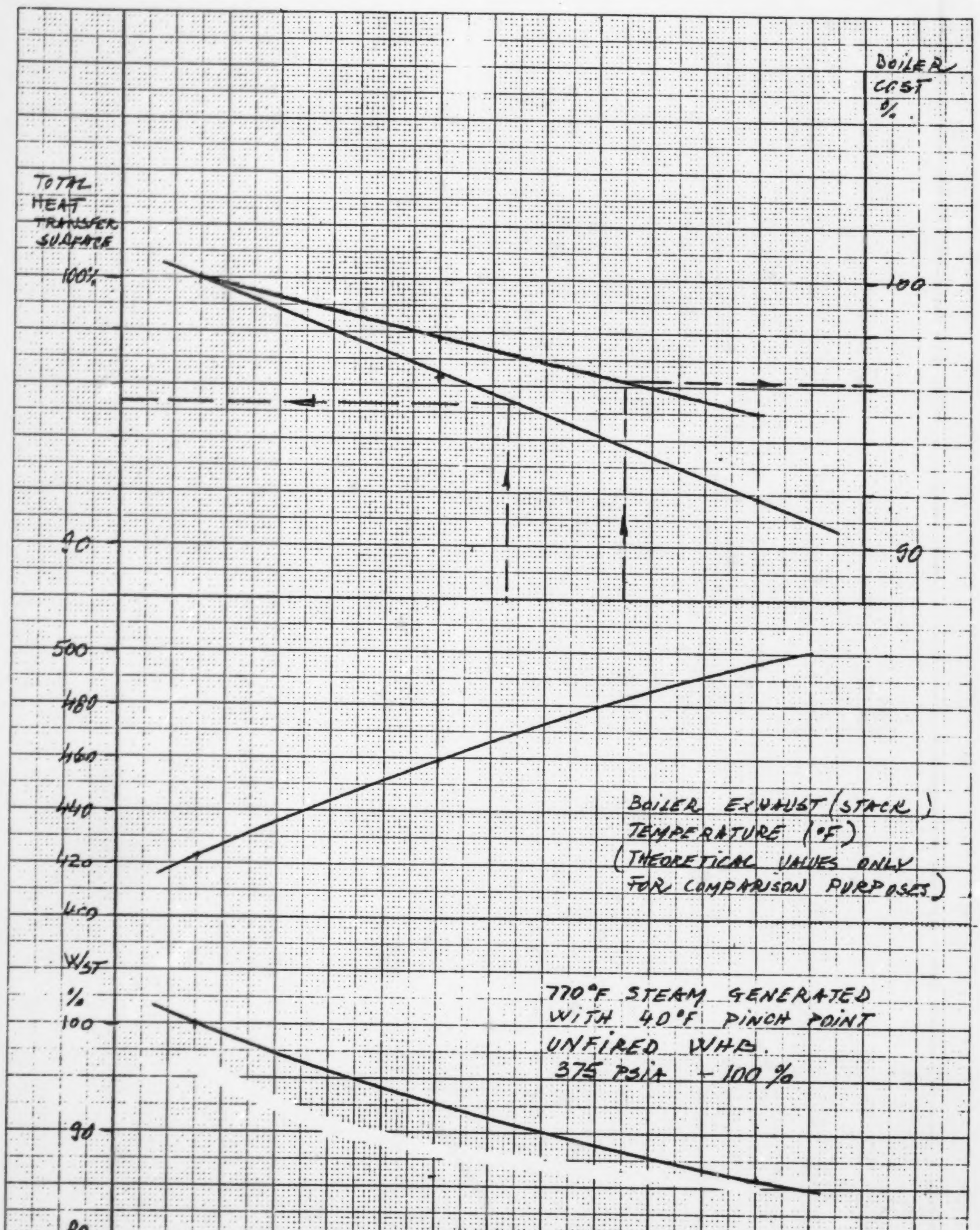
9. Breakdowns, if any, can be repaired at sea by the ship's crew. Any tube leaking can be cut and plugged in outside of the boiler casing. Complete tube replacements can be deleted as long as necessary without having to shut down the whole boiler.

I hope you will find the above information helpful in your investigation. Please do not hesitate to contact us if more information is needed.

Yours very truly,
FOSTER WHEELER LIMITED

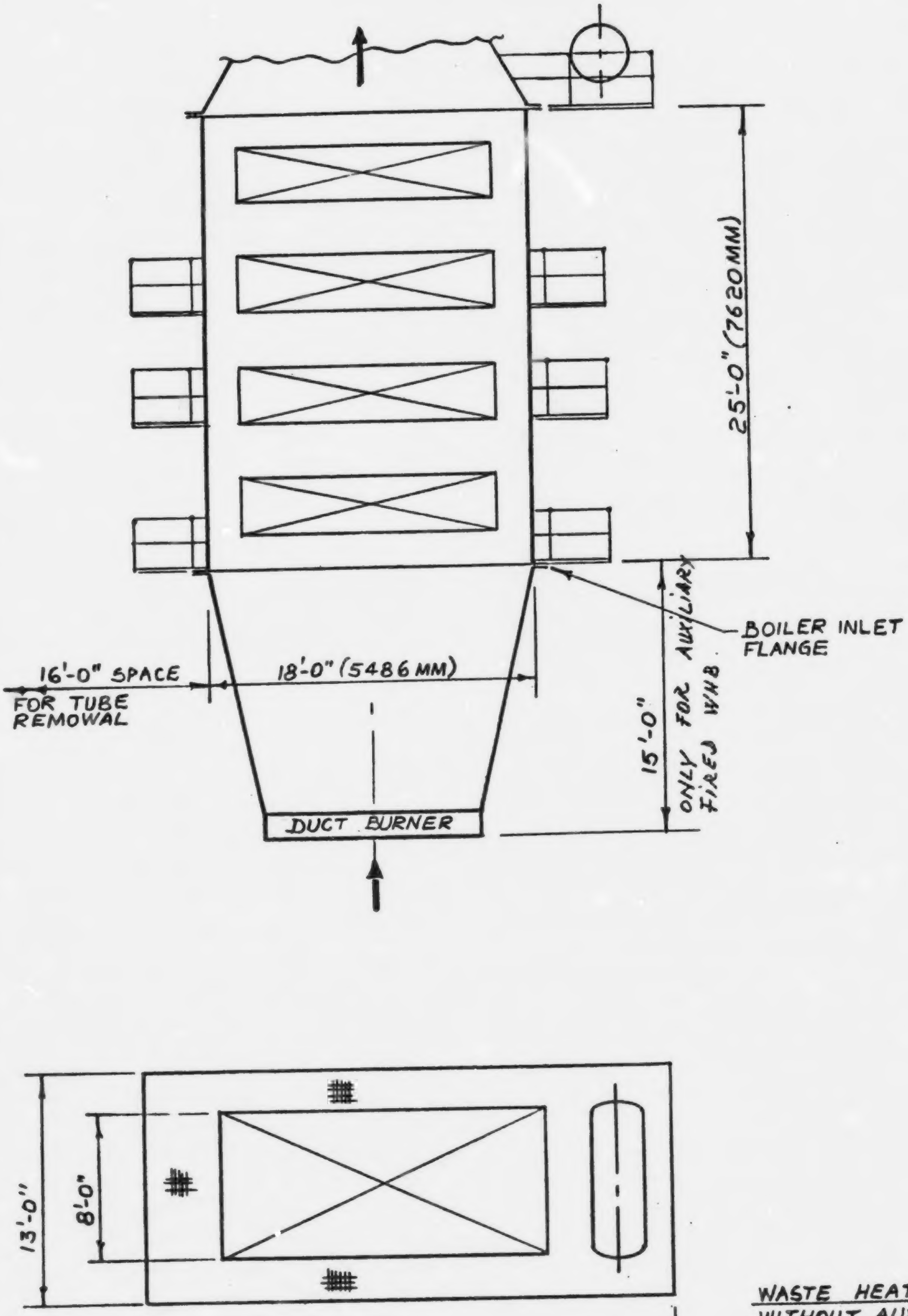
A. Iozsef
Special Products Division

AI/bb
Attach.



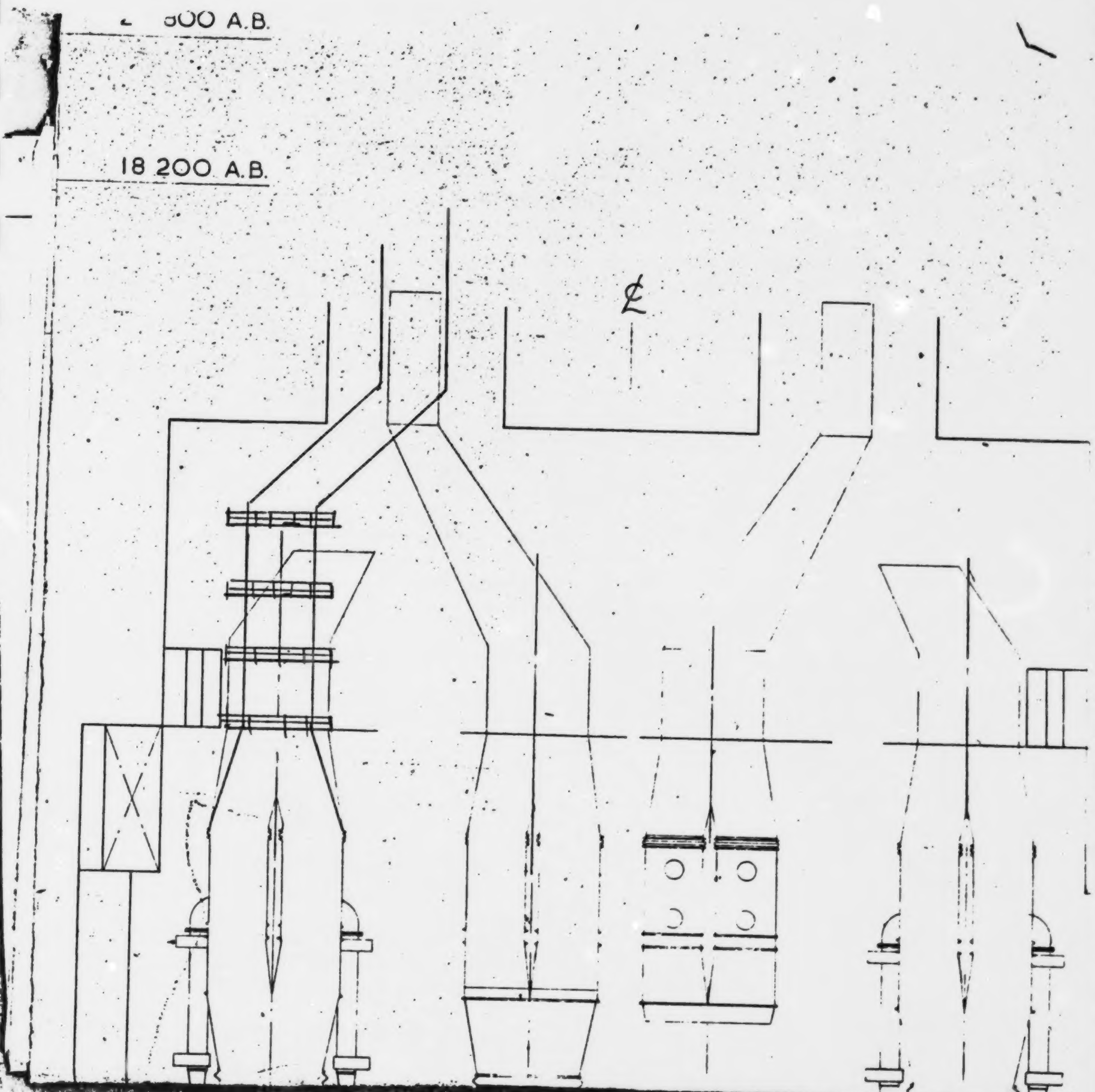
461510
K+E 10 X 10 TO THE CENTIMETER KEUFFEL & ESSER CO. MADE IN U.S.A.

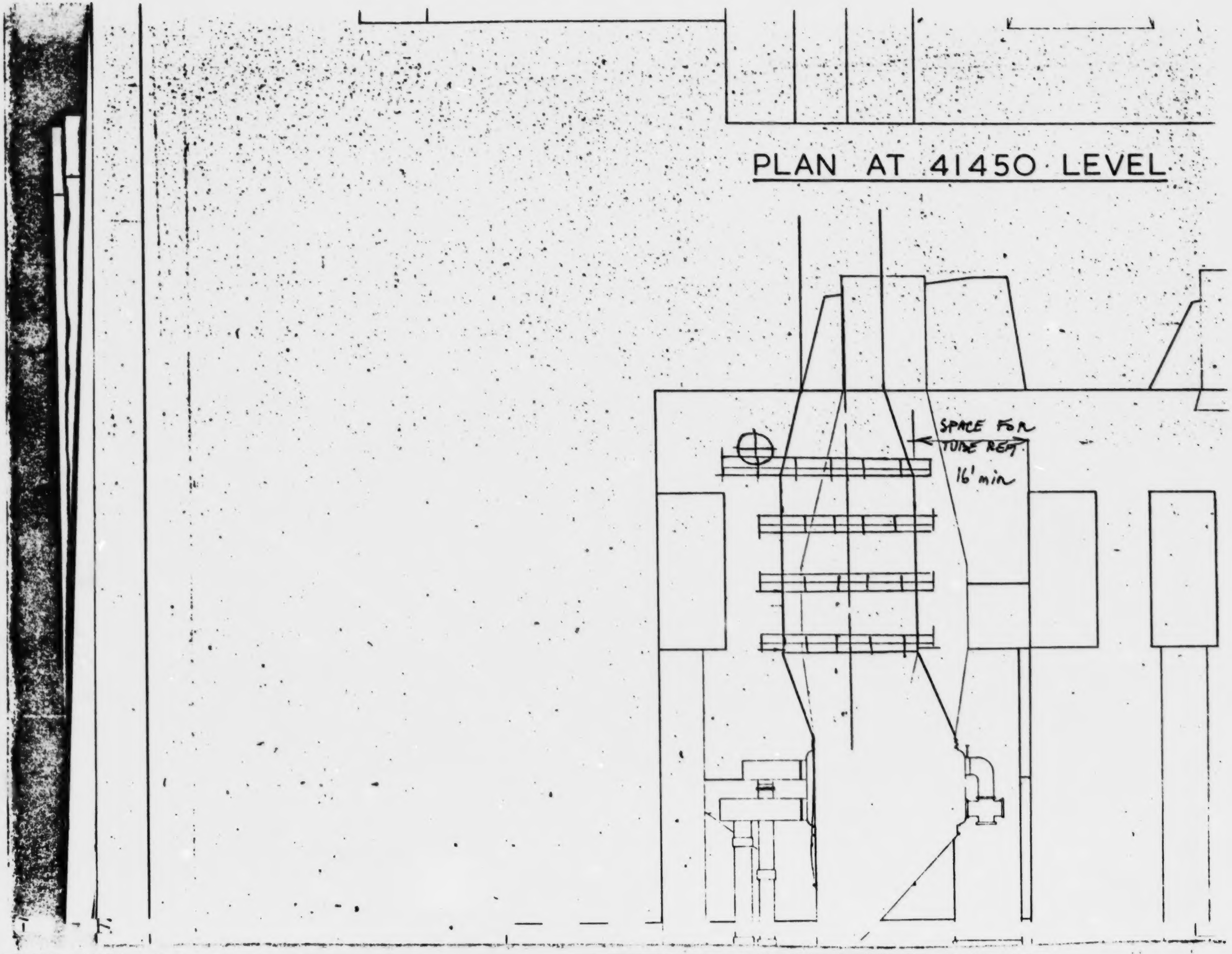
FOSTER WHEELER LTD.


25'-0" (7620MM)
BOILER INLET FLANGE
16'-0" SPACE FOR TUBE REMOWAL
18'-0" (5486 MM)
15'-0"
ONLY FOR AUXILIARY FIRED WHB
DUCT BURNER
13'-0"
8'-0"
29'-0"

WASTE HEAT BOILER WIT.
WITHOUT AUXILIARY
FIRING.

DATE: FEB. 29/80 DWG. CP-801-4


2 800 A.B.
18 200 A.B.
℄


PLAN AT 41450 LEVEL

SPACE FOR TUBE REM'T
16' min

FOSTER WHEELER LIMITED

GAS TURBINE WASTE HEAT BOILERS

REFERENCE LIST

The following is a list of installations of waste heat boilers operating in conjunction with turbine exhaust gases:

UNIT	SUPPLIED BY
West Texas Utilities Company Rio Pecos Station, Giruin, Texas Steam - 360 MPPH, 900 PSIG, 905 FTT Gas Turbine Exhaust - 352 MPPH at 800 Deg. F. Fuel - Natural Gas Supplementary Fired Operation - Since Late 1957	FWEC U.S.
Pacific Petroleum Limited Express, Alberta, Canada Steam - 91 MPPH, 400 PSIG, 550 FTT Turbine Exhaust - 525,000 PPH at 870 Deg. F. Fuel - Natural Gas Supplementary Fired Operation - Since late 1963	FWL Canada
Hawker Siddeley - Orenda Engines Division For R.C.A.F. Stations in Alberta, Manitoba and Sask. Fourteen (14) Units - Supplementary Fired Steam Generator Steam - 7,000 lbs. per hour at 75 PSIG saturated turbine exhaust - 109,440 PPH at 685 Deg.F. Operation since 1961	FWL Canada
Cia. Papelera Maldonada S.A. Monterrey N.L. Mexico One (1) Unit - Supplementary Fired Steam Generator Nat. Gas. 50,000 PPH Steam at 150 PSIG saturated 540,000 PPH gas turbine exhaust at 680 deg.F. Operation since October 1964 (FORCED CIRCULATION)	FWL Canada
Panama Canal Company Canal Zone One (1) Unit - combined cylce steam generator - suppl. fired - Bunker "C" oil. Steam - 200 MPPH at 890 PSI, 905 FTT turbine exhaust - 741 MPPH at 741 Deg. F. Operation since December 1964	FWEC U.S.
Hawker-Siddeley For Consolidated Mining and Smelting Pine Point, Northwest Territory One (1) Unit - Supplementary fired steam generator Nat. Gas. 20,000 PPH steam at 50 PSIG saturated 100,000 PPH Turbine exhaust at 930 Deg. F. Operation since February 1965	FWL Canada

Atlos Hornos de Mexico
Monclova, Coah., Mexico
FWEC
U.S.

One (1) Unit - Supplementary fired steam generator -
Nat. Gas. 50,000 PPH Steam, 450 PSIG, 700 FTT
990 MPPH Turbine Exhaust at 550 Deg. F.
Operation - Since 1967

Lummus Company for Nebraska - Illinois Gas
Ammonia Plant
FWEC
U.S.

One (1) Unit - combined cycle steam generator -
Supplementary fired Nat. Gas. Steam 55,000 PPH,
475 PSIG, 740 FTT. Exhaust - 92,000 PPH at 850
Deg.F. (total from two turbines)
Operation since May-June 1965

Shell Oil Company,
Deer Park, Texas
FWEC
U.S.

Two (2) Units supplementary fired - Nat. Gas
58,000 PPH Steam, 650 PSIG, 750 FTT
715,000 PPH at 830 Deg. F. Turbine Exhaust
Operation since April 1966

Southern California Edison Company
For The Garden State Paper Company
Pomona, California
FWEC
U.S.

One (1) Unit - Supplementary Fired
Fuel - Natural Gas or #6 Oil
Steam 126,000 PPH, 100 PSIG (sat. temp.)
Turbine Exhaust 849,000 PPH at 782 Deg. F.
Operation since Septemebr 1966

American Oil Company
Texas City, Texas
FWEC
U.S.

One (1) Unit - No supplementary firing
Steam 75,000 PPH, 480 PSIG at 700 Deg. FTT
Turbine Exhaust 637,000 PPH at 935 Deg. F.
Operation since January 1968

Huble Oil & Refining Company
Baton Rouge, Louisiana
FWEC
U.S.

One (1) Unit - Supplementary Fired
Fuel - Natural Gas
Steam - 130,000 PPH at 600 PSIG 600 Deg. FTT
Turbine Exhaust - 357,000 PPH at 850 Deg.F.
Operation since June 1968

Dow Chemical Company, Freeport, Texas	FWEC U.S.
One (1) Unit - Supplementary Fired - Nat. Gas 150,000#/hr. Steam 625 PSIG, 750 Deg. FTT 685,000/hr. Turbine Exhaust Gas at 956 Deg. F.	
Shell Curacao, N.V. Curacao, Netherlands Antilles	FWEC U.S.
One (1) Unit - Supplementary Fired Fuel - Combination Natural Gas and #2 Oil Steam - 159,000#/hr. 515 PSIG at 800 Deg. FTT Turbine Exhaust - 737,000#/h4. at 930 Deg. F.	
Phelps Dodge Corporation Morenci, Arizona	FWL Canada
Two (2) Units - Supplementary Fired Fuel - Nat. Gas & Bunker "C" Oil Steam - 151,000#/hr. 685 PSIG at 830 Deg. FTT Turbine Exhaust 649,300#/hr. at 955 Deg. F.	
Dow Chemical Sarnia, Ontario	FWL Canada
Two (2) Units - Supplementary Fired Fuel - Natural Gas Steam - 200,000#/hr. 1450 PSIG at 880 Deg. FTT Turbine Exhaust - 1,740,000#/hr. at 950 Deg. F.	
Texaco Trinidad	FWL Canada
Three (3) Units Fuel - Gas Steam - 145,000#/hr. - 450 PSIG at 550 Deg. F. Operation - Since 1965	
Dow Chemical of Canada Ltd. Red Dee, Alberta	FWL Canada
One (1) Unit Steam 97,000 lbs/hr. at 975 Deg. F. Turbine Exhaust 759,100#/hr.	
Dow Chemical of Canada Sarnia, Ontario	FWL Canada
One (1) Unit - Supplementary Fired, Natural Gas (Future #2 Oil) 540,000 PPH, 1450 psig, 880°F. Steam. Turbine Exhaust 2,268,000 PPH at 960°F (fired to 1400°F) in operation since June 1977	

Dow Chemical of Canada Fort Saskatchewan, Alberta	FWL Canada

Two (2) Units. Nat. Gas Supplementary Fired
520,000 PPH, 850 psig, 725°F.
Turbine Exhaust 2,030,000 PPH at 980°F.
(fired to 1400°F). 1979

City of Medicine Hat, Alberta	FWL Canada

Two (2) Units. Nat. Gas Supplemntary Fired 162,500 PPH, 600 PSIG, 750°F. Steam generated normal firing. 300,000 PPH, 565 PSIG steam at max. firing cond. Turbine Exhaust 1,173,200 PPH at 935°F.

INTER-OFFICE COMMUNICATION
MEMO

To:
A: FILE

Date: March 5th, 1980

FILE:
DOSSIER:

Copy to:
Copie à:

From:
De: John Wainwright

Subject:
Sujet: CONVERSATION WITH H. DOUGLAS MARRON, PROJECT ENGINEER, U.S. NAVY RACER PROGRAM

The US Navy has engaged General Electric, Solar International and Garrett Manufacturing to design a waste heat boiler for the GE LM2500. The designs will be complete in September, at which time the US Navy will select one plan and construct a land-based proto-type unit. The current schedule calls for combined cycle propulsion to be installed aboard US Navy vessels by 1985.

Marron works in the R & D division of the US Navy. His role is to develop equipment designes and then convince the operational divisions that this equipment should be installed aboard Navy vessels. He is very confident of the combined cycle plant being approved by the operational branch since it does agree with the US Navy's objective of reduction of fuel consumption and since the land-based proto-type plant will have undergone 3,000 - 5,000 hours of extensive testing.

The US Navy will not have any crew members aboard their combined cycle vessels dedicated to steam system maintenance. A "steam ticket" is not required on the crew unless there is an oil-fired boiler installed.

The US Navy has not conducted any studies into the reliability of combined cycle.

The US Navy has installed waste heat boilers aboard their Spruance class vessels. These boilers are poorly designed, poorly located and poorly operated in terms of de-aerating and removing solids from feedwater. However, the problems encountered with these boilers does not deter the US Navy's confidence in combined cycle propulsion for their future vessels. The existing boiler problems can virtually be attributed to the shipyard (Litton) which did not pay sufficient attention to steam system design.

Marron suggested I contact Mack Herder and George Huff of General Electric, Lynn Massachusetts for information regarding the GE Racer program boiler.

INTER-OFFICE COMMUNICATION
MEMO

To:
A: FILE

Date: March 6th, 1980

FILE:
DOSSIER:

Copy to:
Copie à:

From:
De: John Wainwright

Subject:
Sujet: CREW STRUCTURE COMBINED CYCLE PROPULSION

I spoke today with Mr. Jenkins, (613-992-6219) of the Senior Examiner of Engineers Division, Transport Canada to ascertain the requirement for a steam "ticket" on the crew of a ship having combined cycle propulsion. He advised that current regulations require a motor or steam ticket when gas turbines serve as the prime mover. A steam ticket is only required when steam turbines serve as the prime mover. This is the case regardless of the ship's auxiliary steam load, pressure or temperature. With the combined cycle plant, since the majority of power is derived from the gas turbines, a steam "ticket" on the crew structure is not necessary.

A draft refinement to the regulations is now being prepared, and should be completed within the month, which would require the crew of a gas turbine vessel to have a "general ticket" which is essentially a combined motor and steam ticket. A second draft will be completed and circulated in August with the refinement coming into effect in approximately three years.

In any event, a ship provided with a combined cycle propulsion system is not required to have a crew structure which differs in training and numbers from a ship propelled by gas turbines only.

SUN ROD Sunrod ab

Melville Shipping Ltd
La Compagnie de Navigation Melville Ltée
1253 McGill College,Suite 301
MONTREAL, Que. H3B 2Y4
CANADA

Er referens/ Your reference	Vår referens/ Our reference	Datum Date
Mr J G Wainwright	J Wester/NÅ	1980-03-11

Gentlemen:

Re: Combined Cycle Propulsion System
Icebreaking LNG Vessels

We refer to your letter of January 25, 1980, regarding waste heat boiler designed for combined cycles. We have with interest studied your letter and have the following comments to your questions:

A. Waste Heat Boiler Design

1. Design

For this installation, we would suggest one exhaust gas boiler to be installed after each gas turbine. The physical dimensions can be seen from enclosed drawing No 3202675. Please note that this drawing indicates steam operated soot blowers, which we do not consider to be a necessity for LNG units.

However, it could be worth while to investigate if the number of steam drums should be decreased to the same number as steam turbines. To do this one must know the operating modes of the different gas turbines.

2. Technical data

Enclosed please find technical data for our proposed design which, in addition to your preliminary specification includes a unit for de-airating feed heater, which eliminate the need for steam extraction from turbine to de-airator.

../

The pinch point and approach temperature you have selected, i.e. 40°F and 80°F, is the same figures as we would recommend for practical use today. The steam pressure in the superheater outlet must be optimized together with the steam turbine manufacturer, bearing in mind that the generated steam flow will decrease when the steam pressure increases.

3. Supplementary firing

Supplementary firing could easily be arranged for generating an additional 20,000 lbs steam/h. This supplementary firing will increase the exhaust gas temperature before boiler to 1,000°F. This means that the supplementary firing cannot be used when the gas turbine operates at high temperature levels.

In order to produce more steam by supplementary firing, a muc more complicated and expensive design of the exhaust gas boiler would be necessary, i.e. water cooled furnace.

4. Dry firing

By selecting low alloy steels, the boiler can accept maximum 1000°F at dry conditions during short periods of time. If th gas temperature will be exceeded, we recommend a by-pass arrangement of the boiler.

B. Waste Heat Boiler Control

1. Control

As the waste heat boiler generates steam depending only on the exhaust gas flow and temperature, basically two concepts for control can be used:

a) the steam turbine acts as a pressure controller and keeps the steam pressure in the WHB constant by changing loads

b) the steam turbine throttle can be locked in an open position and steam pressure in the system will vary according to the actual conditions.

The control of the steam turbine is preferably connected to the ship's manouever system during transient conditions.

If the steam turbine cannot consume all of the generated ste , an automatic dumping valve should be installed to by-pass steam direct to the condenser.

2. Start up procedure of WHB

The start up sequence of the boiler is very simple and could be interlocked into the starting sequence of the gas turbine Components to be included in this interlock will be circulat g pumps and feed water pump.

../

3. Ice-ramming

During rapid transients, problems with water level in the steam drum will occur, due to rapid changes in water content of the evaporator section. The best and most safe way to solve this, is to oversize steam drums to be able to absorb the total evaporator volume in the steam or water spaces of the drum.

At 2500 hp/gas turbine the exhaust gas boiler will produce approximately 30% of design steam quantity. We do not think that 1,000 ice-ramming routines per year will make any difficulties for the boiler with respect to fatigue stress problems. However, this type of operation will be dangerous for the circulating pump. To overcome this, the design of the circulating pump is very important, especially with respect to the NPSH, as well as suction pipe from steam drum to circulating pump.

C. Reliability

1. We regard the circulating pump to be the most critical component with respect to reliability. The Sunrod boiler is mostly used on board diesel ships for waste heat recovery. We have also delivered land based boilers for waste heat recovery in the process industry. Our experiences tells us that the heat exhaust unit have practically no failures at all during operation. The few cases we know about was caused by restricted circulating water flow.

D. Future Waste Heat Boiler Improvements

For this type of application we cannot today forsee any design improvements likely to occur in the near future.

We hope that this letter will give you sufficient answers to your questions. We are very interested to continue to co-operate with you to find out the best design of waste heat boiler system for your ship.

If you need any further information, please contact us.

Very truly yours,

SUNROD INTERNATIONAL AB

Jan Wester

c.c. Fairwater Sales Ltd, Mr Al Krauth

BOILER DESIGN

The basic design of the proposed Sunrod boiler is shown on drawing No 3202675.

The Sunrod extended surface consists of plain tubes with steel rods of good heat-transmission properties fusion welded to the tubes. The tube itself has an outer diameter of 48.3 mm and a wall thickness of 4.5 mm. The welding is executed in automatic welding machines.

The Sunrod tubes are placed in a staggered pattern to obtain maximum heat transfer. The pitch (longitudinal) of the rods is 15 mm. The relatively long distance between the rods assures that normal soot deposits will hardly effect the pressure drop on the gas side.

The tubes are welded to the end support plates. At one end (the header end) the support plates are fixed to the casing but in the other end they can move to allow free expansion.

The design is sturdy to endure heavy vibrations occuring in a ship operating in ice.

The superheater consists of plain tubes with an outside diameter of 33.7 mm and a wall thickness of 3.6 mm, arranged in a staggered pattern.

TECHNICAL DATA

Boiler

Steam capacity	lbs/h	53 000
Steam temperature	°F	770
Steam pressure	PSIG	355
Feed water temperature (DAH) (Condensate return temp.)	°F	110
Exhaust gas flow	lbs/h	558 000
Exhaust gas temperature inlet	°F	850
"- outlet	°F	342
Total draft loss	in	7"

Pumps (at above load point)

ECO-DAH water flow	lb/h	100 300
Pressure drop (heating surface)	PSI	27
Evaporater water flow	lb/h	160 000
Pressure drop (heating surface)	PSI	16

AUTOMATIC CONTROL SYSTEM

The boiler is equipped with an automatic control system including electronic transmitters and electrical control valves.

All of the functions are built-in in one control cubicle which is to be located in the control room for remote operation of the heat recovery boiler system.

High pressure drum

- A feed water level control with feed forward signal from the steam flow transmitter.

- A water level indicator with built-in contacts for high and low level alarm.

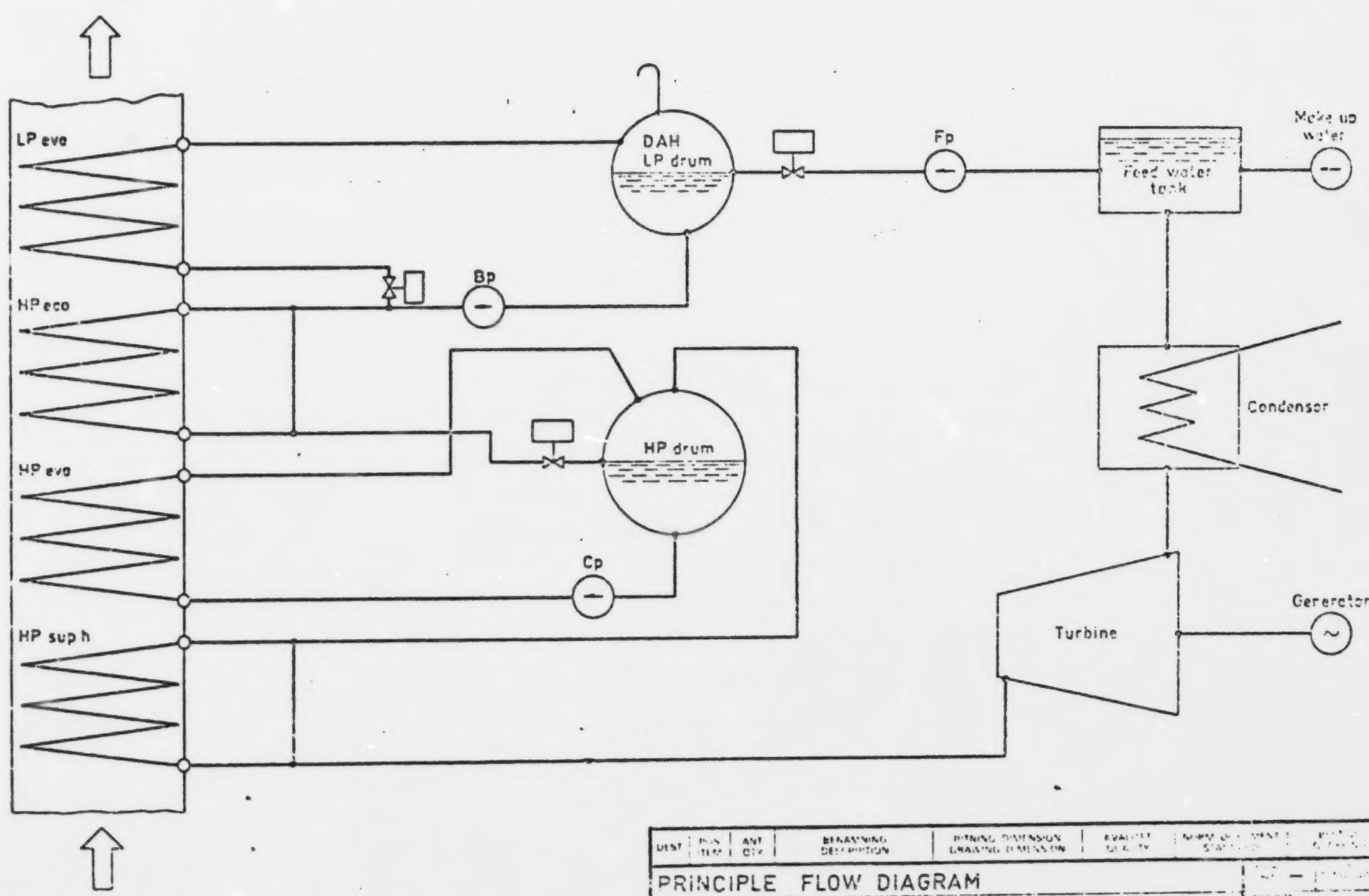
- A steam pressure transmitter with indicator and alarm.

De-airating feed water heater

- A feed water control valve which controls the setting of the overflow valve and make-up feed valve.

- A water level indicator with built-in contacts for high and low level alarm.

- A steam pressure control which controls the make-up steam to the DAH, and alarm.

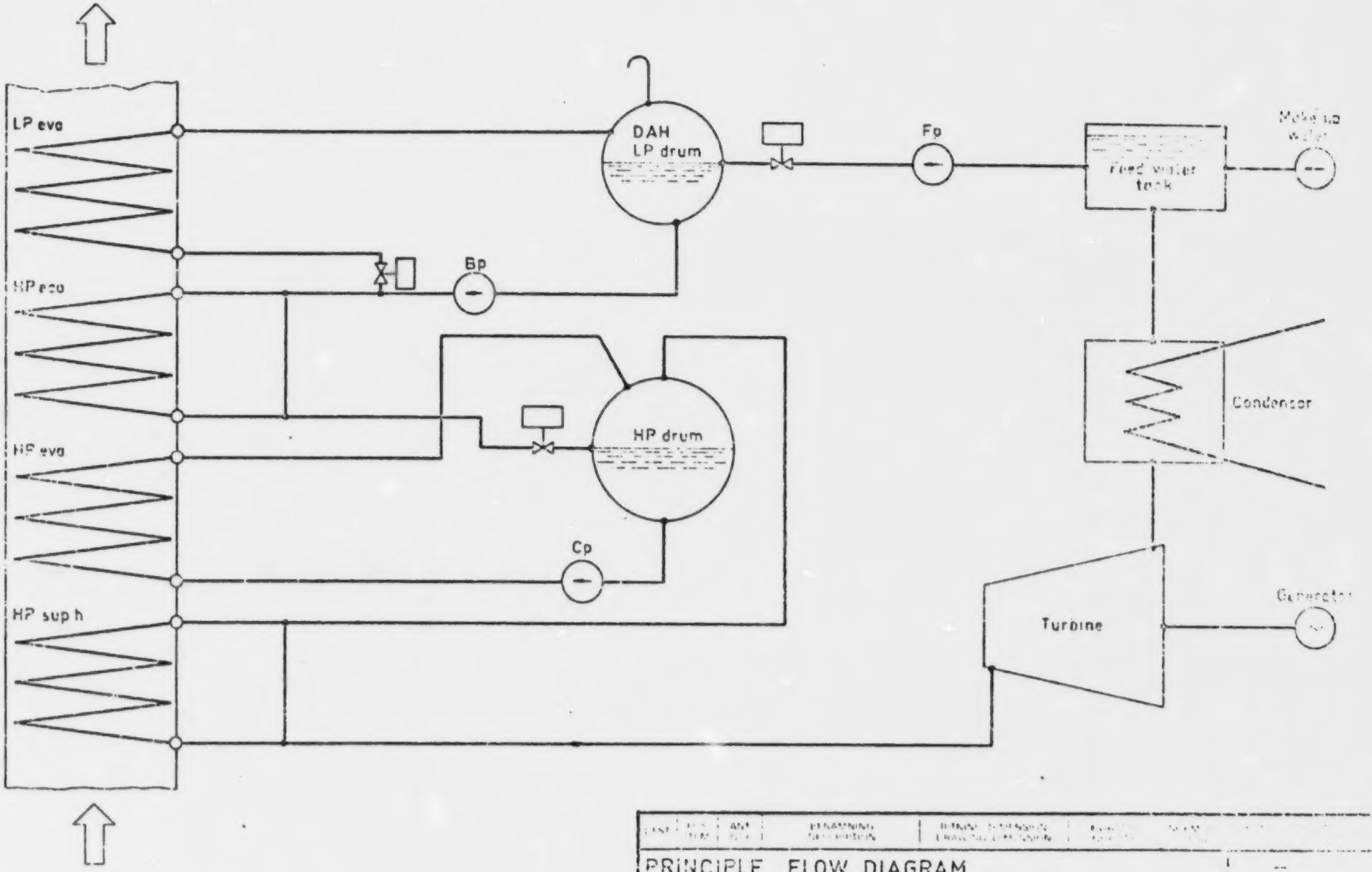

LP eva
HP eco
HP eva
HP sup h
DAH
LP drum
HP drum
Bp
Cp
Fp
Feed water tank
Make up water
Condenser
Turbine
Generator

DEST	POS ITEM	ANT QTY	BENÄMNING DESCRIPTION	RITNING DIMENSION DRAWING DIMENSION	KVALITET QUALITY	NORM STANDARD

PRINCIPLE FLOW DIAGRAM
COMBINED. CYCLE
ALT. 5.1.C

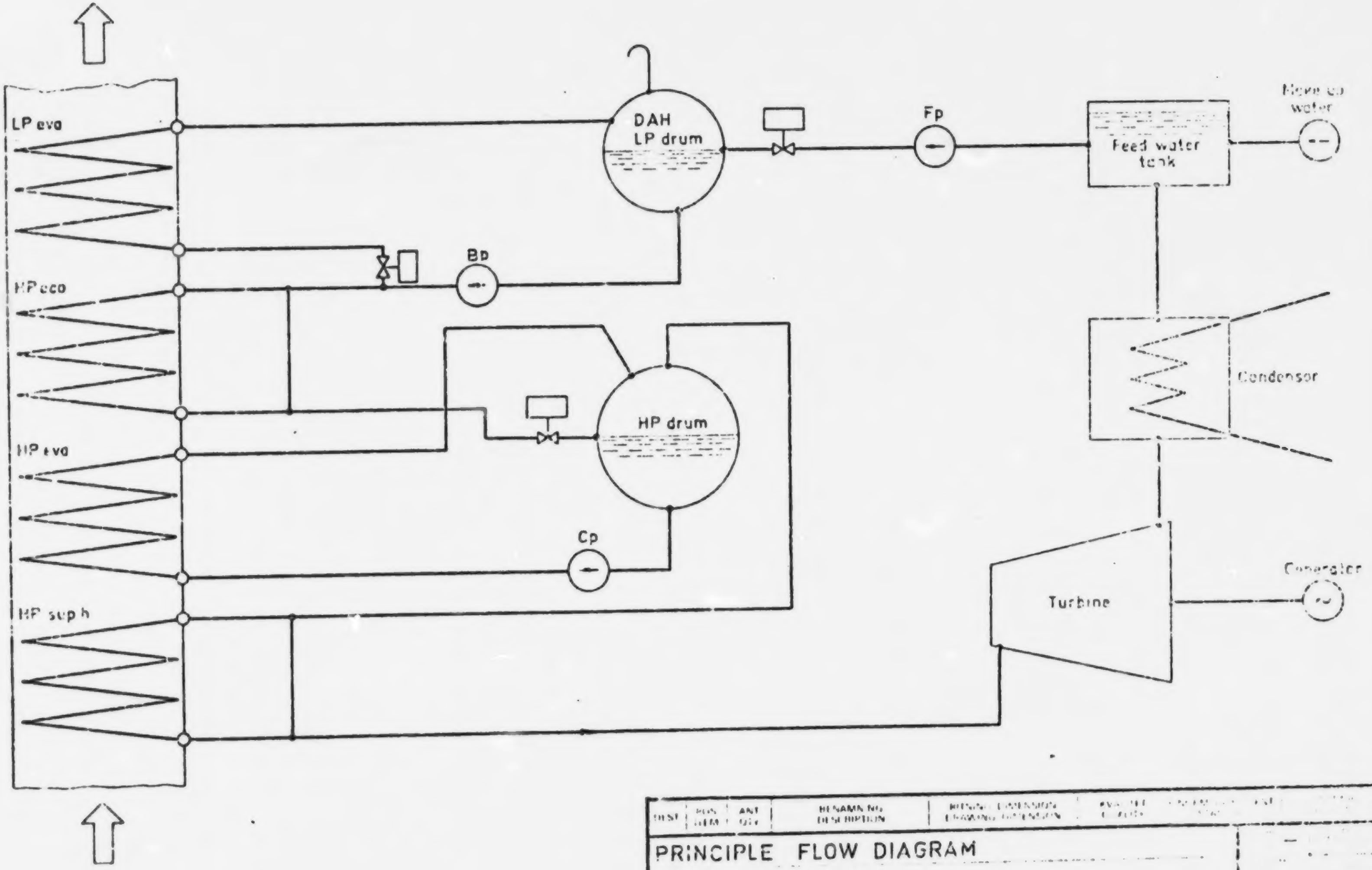
L. Elmon 77-03-28

8107


LP eva
HP eco
HP eva
HP sup h
DAH
LP drum
HP drum
Bp
Cp
Fp
Feed water tank
Make up water
Condensor
Turbine
Generator

PRINCIPLE FLOW DIAGRAM
COMBINED CYCLE
ALT. 5.1.C

L. Elmon [illegible]-03-28


LP eva
HP eco
HP eva
HP sup h
DAH
LP drum
HP drum
Bp
Cp
Fp
Feed water tank
Make up water
Condensor
Turbine
Generator

DEST	POS ITEM	ANT QTY	BENÄMNING DESCRIPTION	RITNING DIMENSION DRAWING DIMENSION

PRINCIPLE FLOW DIAGRAM
COMBINED CYCLE
ALT. 5.1.C

L Ekman 77-03-28

GEC Turbine Generators Limited

Member of the GEC Power Engineering Group

NAVAL DEPARTMENT
INDUSTRIAL AND MARINE
STEAM TURBINE DIVISION

Willans Works, Newbold Road, Rugby,
Warwickshire CV21 2NH
Telephone Rugby (0788) 77111 ext: 559
Telex: 31463
Telegrams: Enelectico Rugby Telex

J G Wainwright, Esq.,

Project Engineer,
Melville Shipping Limited,
La Compagnie de Navigation Melville Ltee,
1253 McGill College, Suite 301,
Montreal Quebec, Canada. H3B 2Y4

Your ref:

Our ref: ND/DGN/MKG

19th March 1980

Dear Mr Wainwright,

Thermal Cycling of Steam Turbines

Peter Donnison has sent me a copy of your letter on this subject dated March 3rd. It was most interesting to receive this letter as, whilst it is an aspect of combined steam and gas turbine operation which we have given consideration to, we have never seen any reference to it in either technical papers or technical presentations on the subject.

We have been interested in the potential application of combined cycle propulsion plant for many years and have investigated the thermal stresses imposed on the steam turbine as a result of the variation in gas turbine exhaust temperature with load and hence the variation in steam temperature with gas turbine output. I enclose an illustration, Figure 1, showing a typical variation of steam temperature with gas turbine load and you can see a variation of about 140°C in steam temperature when the gas turbine load changes from 100% to 50%. We are able to calculate the resulting thermal stresses induced in the steam turbine for varying times over which this temperature change takes place and the most critical component to be considered is the steam turbine rotor at the position of the HP gland where the temperatures are highest and high heat transfer co-efficients apply.

Another illustration is included Figure 2, which illustrates the significance of turbine rotor diameter to the level of thermal stress experienced for a given rate of temperature change and you will notice that there is a very significant connection between diameter and stress level. In order to apply these calculations it is necessary to possess adequate thermal fatigue data for the materials being considered. We in fact possess very adequate data as a result of experiments carried out in our metallurgical laboratories over a very long period of time so that we now have curves of high strain fatigue plotted against cycles to failure together with curves indicating the number of cycles to crack initiation for various modes of thermal cycling.

The effect upon the fatigue life of the component will depend upon the mode of operation and this is illustrated on Figure 3 which shows a comparison of force cooling with slow cooling.

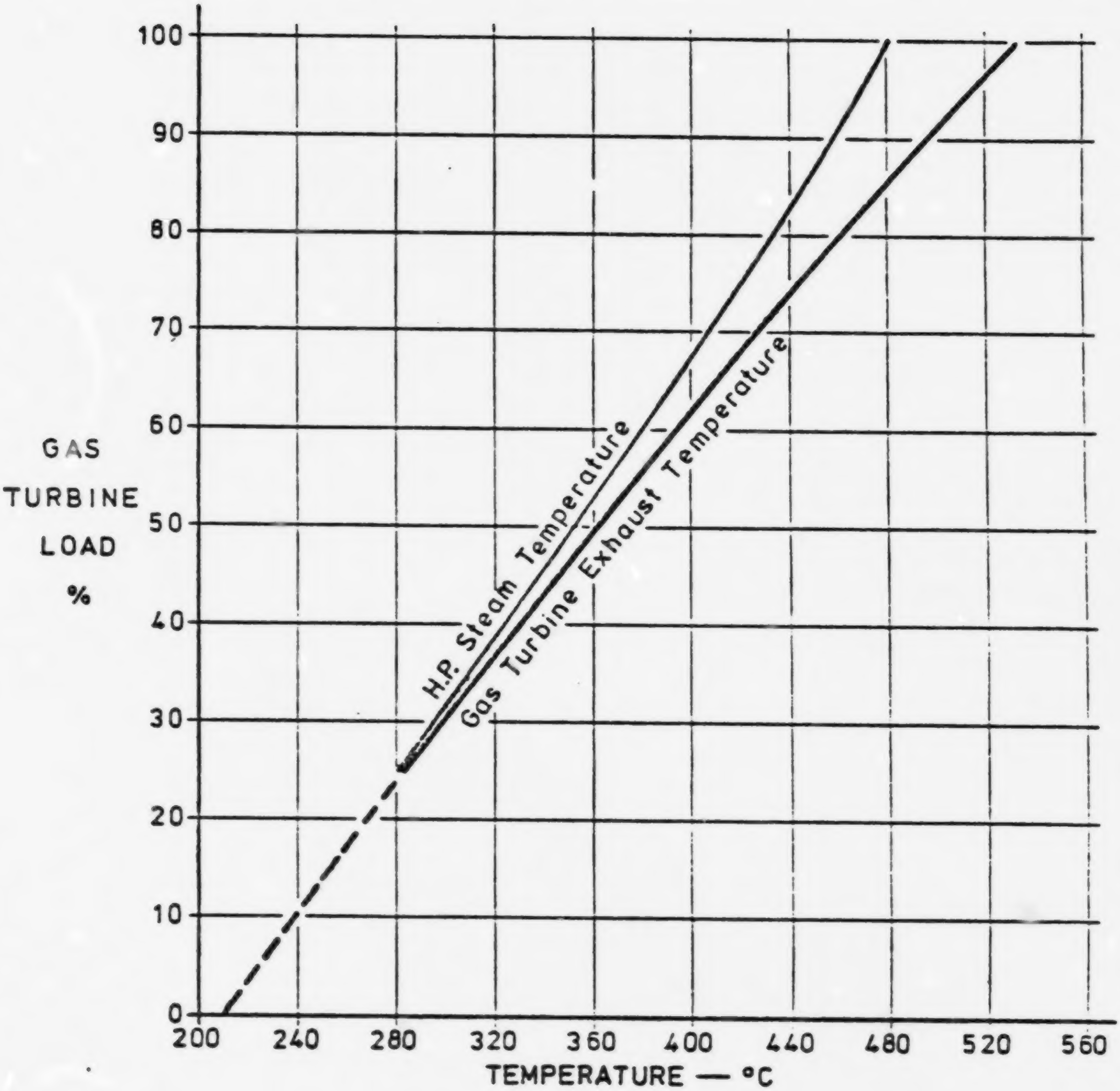
For the application of a combined cycle propulsion plant to an icebreaker I do not think the ice ramming cycles will cause any problems as the cycle times are so short even with the mechanical drive system you discussed with our gear company, the time taken from full ahead to full astern is only 35 seconds before full power, and the associated steam temperature levels are reestablished. Hence the critical rotor surface temperatures will be virtually unaffected and hence thermal stress will be very low.

We are very interested in this project and would appreciate being kept in the picture and we would also be very happy to provide any further information you might require in connection with steam turbines.

Yours sincerely,

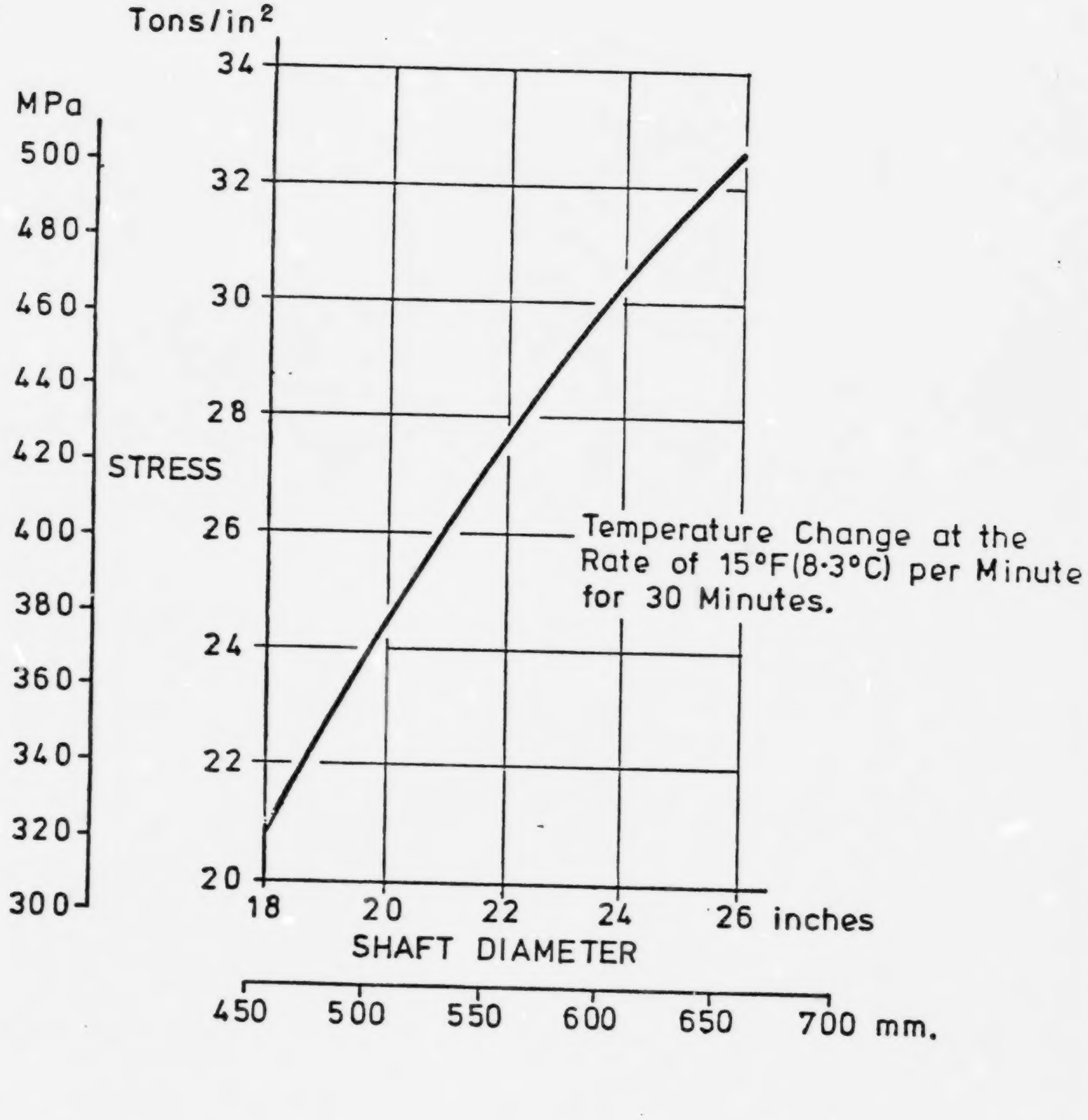
D G Nicholas
Manager

GEC



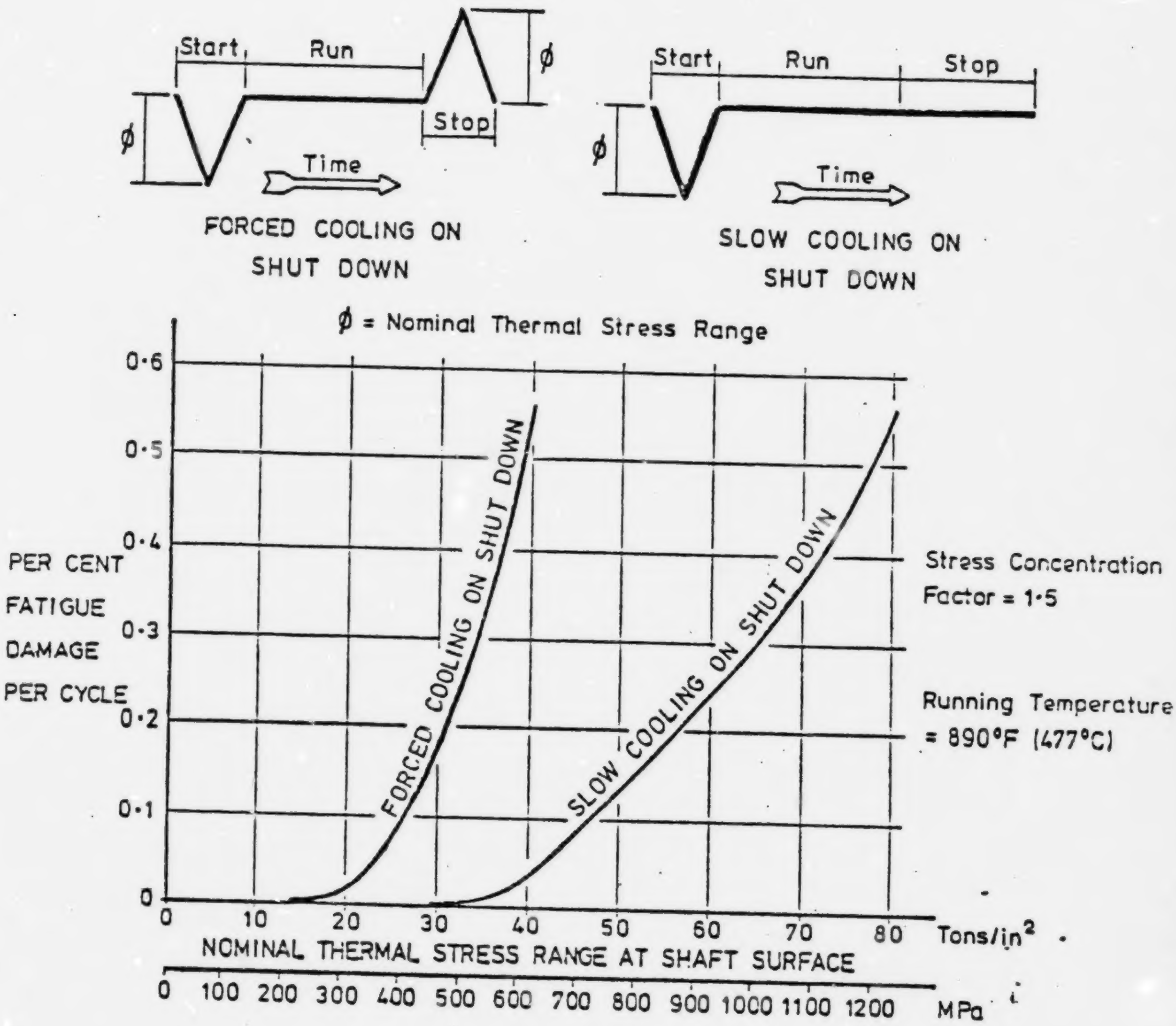
Gas Turbine Load against Gas Turbine Exhaust & H.P. Steam Inlet Temperature.

FIGURE 1



Thermal Stress Against Shaft Diameter

FIGURE 2



Cyclic Fatigue Damage against Thermal Stress Range for Different Modes of Cooling

FIGURE 3

ROLLS-ROYCE (CANADA) LIMITED LIMITÉE

9500 COTE DE LIESSE ROAD
LACHINE, QUEBEC
MAILING ADDRESS:
P.O. BOX 1000, MONTREAL AMF
MONTREAL, QUEBEC, CANADA
H4Y 1B7

PS.310

15 April 1980

Mr. John Wainwright
Project Engineer
Melville Shipping Ltd.
1253 McGill College
Suite 301
Montreal, Quebec
Canada H3B 2Y4

Dear John:

Rolls-Royce Combined Cycle Experience

The following is in answer to your request for information on the operating experience of our units in combined cycle.

To begin with, none of the three installations currently operating are true combined cycle in the sense that steam is produced in the waste heat recovery unit and used to drive a steam turbine. Our experience to date has been confined to using waste heat recovery for the production of steam for process purposes or the utilization of exhaust gas for other industrial processes in shore-based applications. The three installations are: Esso Refinery at Milford Haven, South Wales; Shell Chemicals at Carrington, England; and Shell Canada Resources at Waterton, Alberta. The first two utilize Avon gas generators and the third is RB.211. These installations have all operated successfully with no significant problems.

Details of each installation are as follows:

Esso Refinery - Milford Haven

The Avon gas generator drives a power turbine which in turn drives an AC generator. Exhaust gas from the gas turbine is ducted to a furnace where it is used as the primary air for supplementary burners which add further heat. The hot gas exiting this furnace is used in the crude oil cracking process.

Commissioning - 1968

Electrical output - 9.9 Mw

Running time - 79,650 hours (as of June, 1979)

Availability and Reliability - greater than 95%

NOTE: Availability and Reliability are defined as follows:

$$\text{Availability} = \frac{\text{Total possible time-inspection/change out time}}{\text{Total possible time}}$$

$$\text{Reliability} = \frac{\text{Total possible time-unscheduled outage}}{\text{Total possible time}}$$

Shell Chemicals - Carrington
As at Milford Haven, the gas turbine (Avon gas generator) drives an AC generator. Exhaust gas passes through a waste heat boiler. The steam generated in this boiler is used in one of the plant chemical processes.

Commissioning - 1968

Electrical Output - 10.3 Mw

Running time - 45,400 hours (as of June, 1979)

Availability and Reliability - greater than 96%

Shell Resources - Waterton, Alberta
The RB.211 gas generator is mated to a Dresser Clark power turbine to drive a natural gas compressor. Exhaust gas is taken to a waste heat recovery unit which uses three separate coil systems to heat oil for a hot oil circulation system and two natural gas streams for the regeneration of a gas dehydrator and a propane treater.

Commissioning - September, 1978

Output - 26,400 hp to natural gas compressor

Running time - 8,940 hours (as of October, 1979)

Availability and Reliability - approximately 98%

We have recently been awarded a contract from Holland for two SK-30 generating sets in combined cycle. Obviously it is too early to comment on the service experience of these units, but they will be adding to our combined cycle knowledge in the years ahead.

Yours very truly,

Bruce B. Tedford
Marketing Representative

:iw
Att.

SEP - 9 1980

SOLAR TURBINES INTERNATIONAL

September 3, 1980

Mr. John G. Wainwright
Arctic Pilot Project
401 9th Avenue, S.W.
675 Gulf Canada Square
Calgary, Alberta, Canada

Dear John:

As you requested we have made a computer simulation run in order to predict the response of our combined cycle plant to the icebreaking duty cycle. We feel that the results are very encouraging.

In an attempt to test the boiler response to the maximum, we used step function input changes and a feed forward command only. At time zero the gas flow was reduced from 100% to 33% mass flow and from 829°F to 659°F. This resulted in a heat input reduction to 28% of steady state. The feedwater flow was also reduced from 100% to 20% at time zero. At time fifteen seconds the steady state values were reinstated.

The attached graph shows the outlet steam flow as a function of time. I have plotted the total steam flow as a percent of its steady state value versus time. This steam flow represents the summation of the high and low pressure flows. We do not feel that the results show too great a thermal shock. In fact, the location of the dry out zone moves only a total of one tube during the entire cycle.

If we actually run this test, the results will be much more stable. The gas turbine cannot provide a step function response in either direction, nor can the feedwater control valve be changed so abruptly. Depending on the results desired, the control system would be modified to aid the goals. For instance, we could anticipate the gas turbine exhaust conditions or we can dump steam to the condenser, or both.

The most encouraging result is that the boiler response indicates thermal shock is not as bad as anticipated and that the system responds well in cycles even this extreme.

Please do not hesitate to contact me to discuss this or any other subject.

Sincerely yours,

G. A. Clough

GAC:mn
Attachment

An Operating Group of International Harvester

2200 Pacific Hwy., P.O. Box 80966, San Diego, CA 92138
TEL (714) 238-5500, TLX 695045, TWX 9103352090

SAOI 2795

SOLAR
DIVISION OF INTERNATIONAL HARVESTER

SHEET NO. ________ OF ________

FOR JOHN WAINWRIGHT MELVILLE SHIPPING

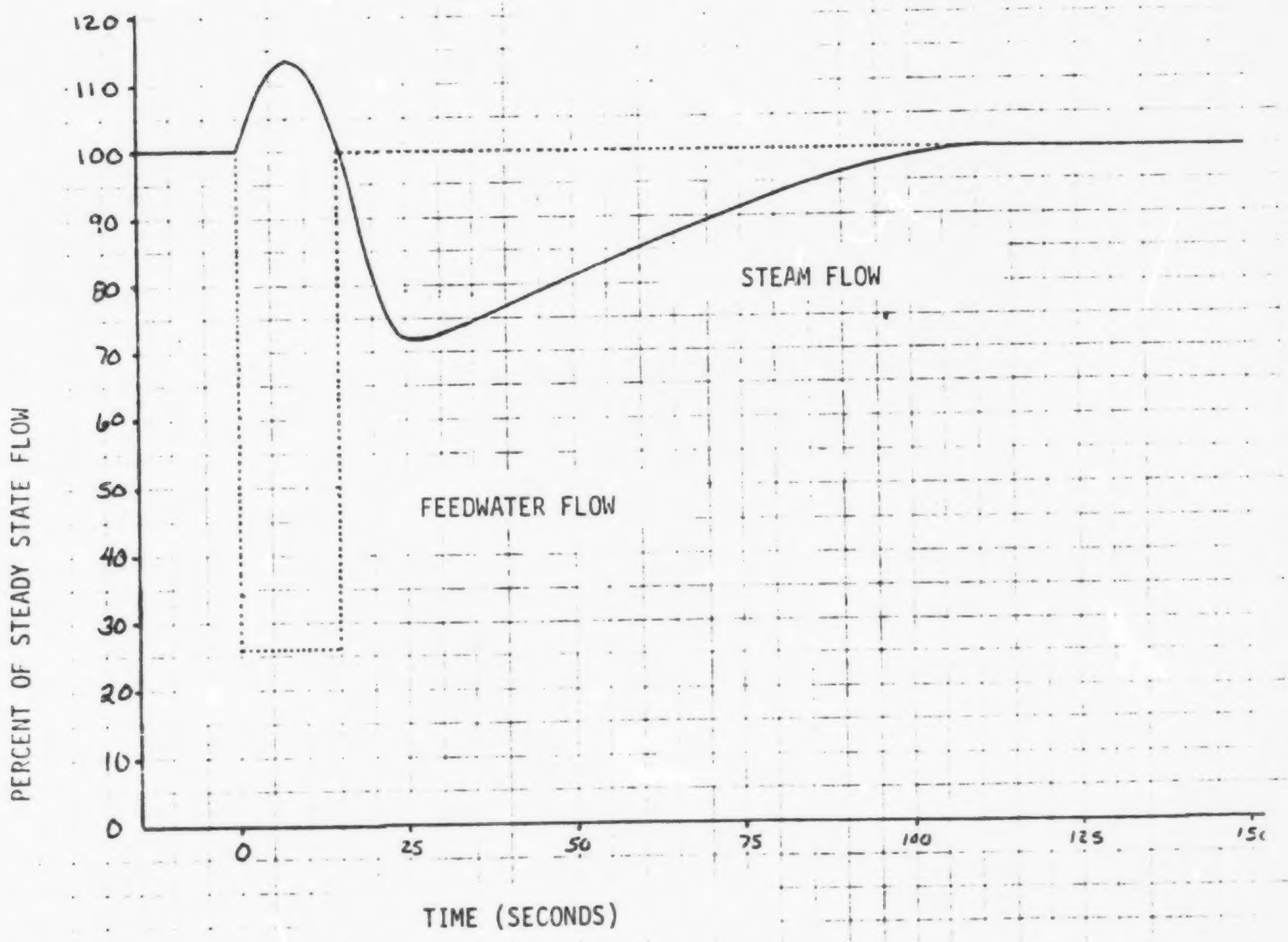
SUBJECT WASTE HEAT BOILER RESPONSE

BY CLOUGH DATE 28 AUGUST 1980 REFERENCE

NOTE: 1 - Gas side and feedwater flow were input as step functions simultaneously.

2 - Used simple feed forward command.

3 - Used no trim control.



SOLAR

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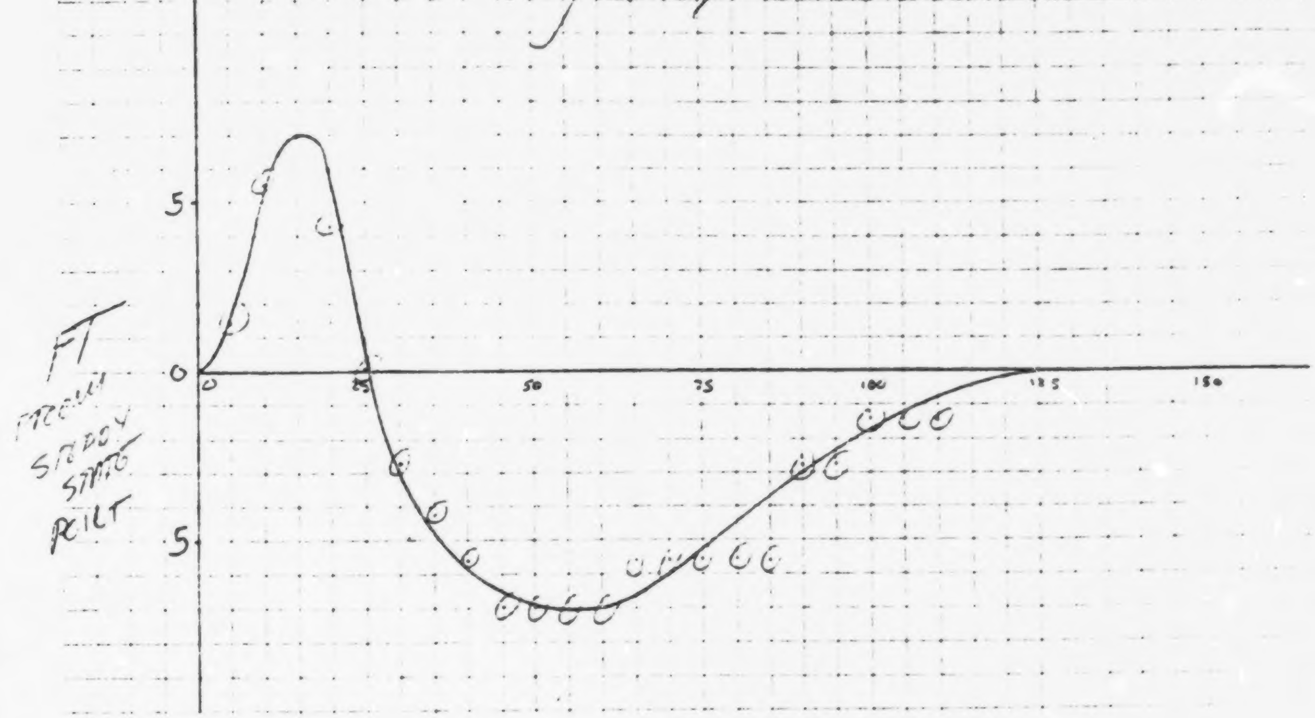
SUBJECT Waste Heat Boiler Response

BY Clough DATE 28 August 1980 REFERENCE

Location of Dryout Zone



SADI 2795

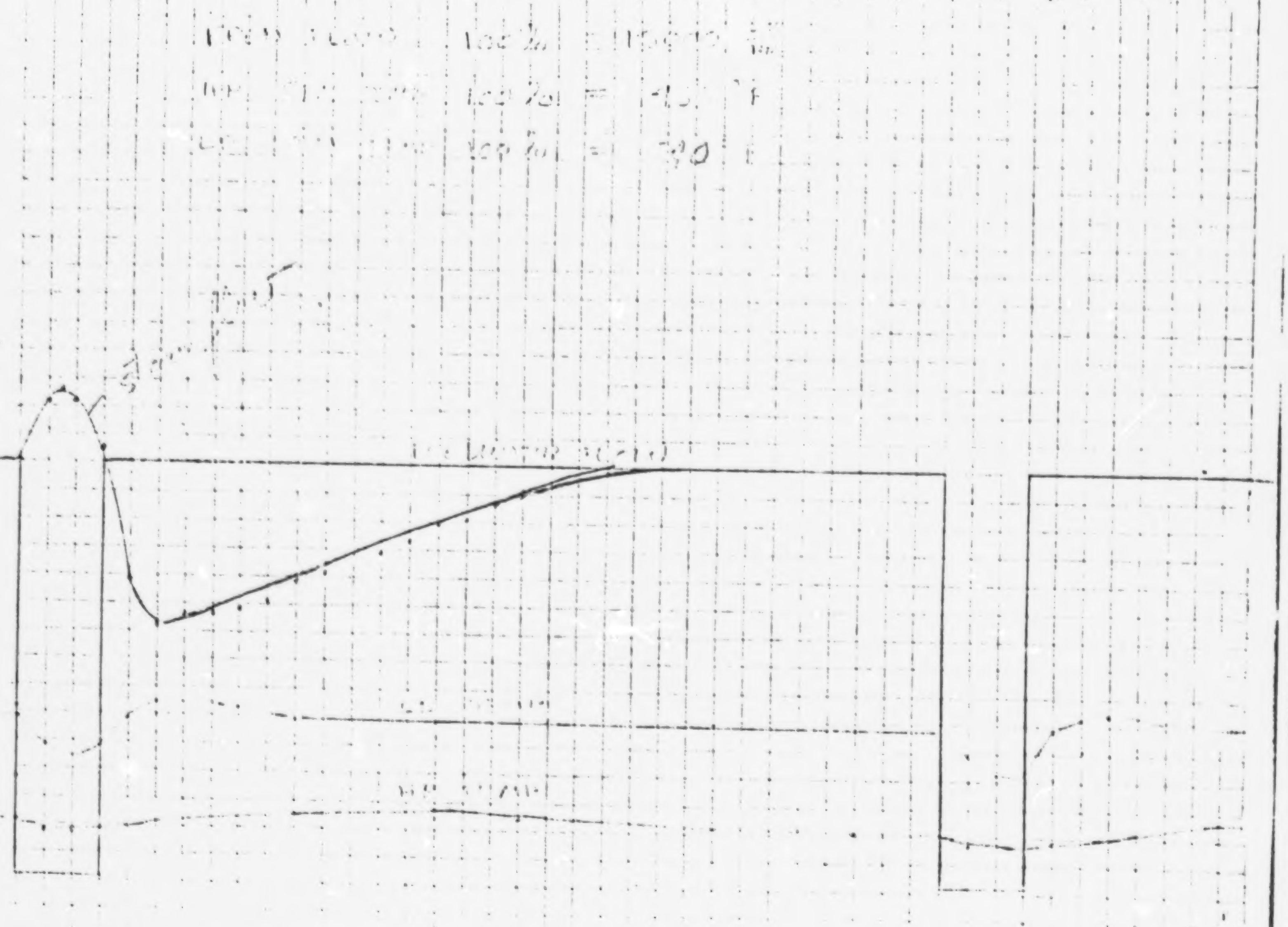
SOLAR
DIVISION OF INTERNATIONAL HARVESTER

SHEET NO. ______ OF ______

FOR ______

SUBJECT ______

BY ______ DATE ______ REFERENCE ______



Temperatures

TIME	H	L
0	[illegible]	~~[illegible]~~
—	[illegible]	415
10	[illegible]	412.5
15	[illegible]	410
20	345	[illegible]
25	[illegible]	[illegible]
30	[illegible]	370
35	[illegible]	370
40	[illegible]	[illegible]
45		
50	375	[illegible]
55		
60		
65		
70	/	
75	[illegible]	340.0
80		
85		
90		
95		
100		
105	[illegible]	[illegible]
110		
115		
120		
125	[illegible]	[illegible]
130		

TEMPERATURES (CONT'D)

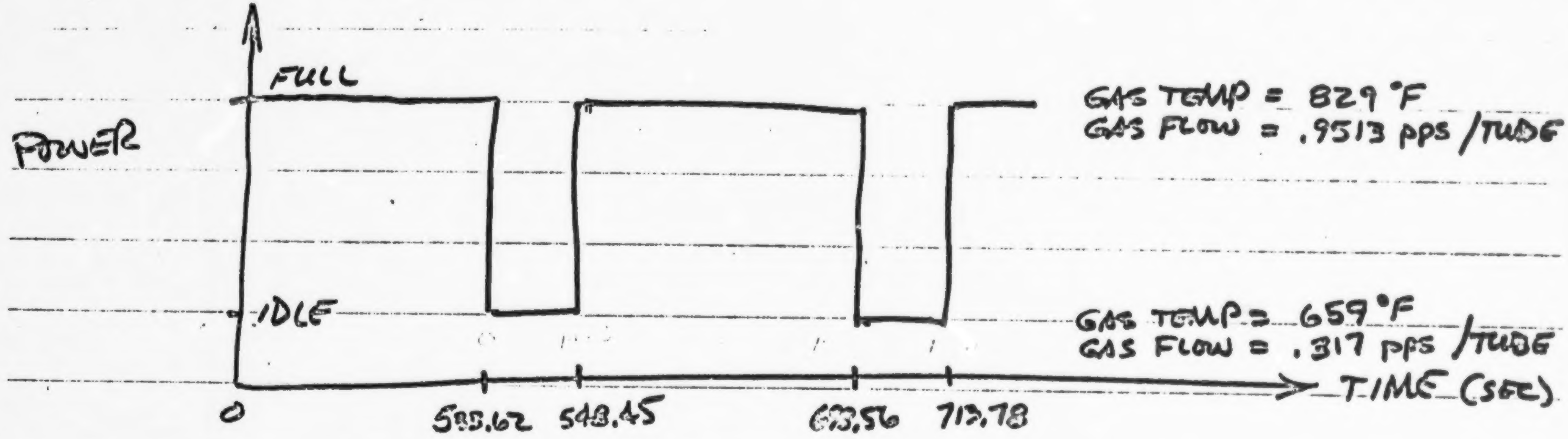
135		
140		
[illegible]		
150	[illegible]	[illegible]
155		
160		
165	[illegible]	[illegible]
170	[illegible]	477.5
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
185	[illegible]	[illegible]
190	[illegible]	[illegible]
195	[illegible]	[illegible]
200	[illegible]	[illegible]
205	[illegible]	[illegible]
210	[illegible]	[illegible]
215	[illegible]	[illegible]
220	[illegible]	2
[illegible]	[illegible]	517.5
330	[illegible]	[illegible]
[illegible]		
245	[illegible]	[illegible]
[illegible]	375.0	[illegible]
[illegible]	367.5	[illegible]
300	360.0	[illegible]

DRY OUT ZONE

Sec	LP Loc	FT		HP Loc	FT	
0	62	43.4		57	79.8	0
5	75	52.5	+9.1	58	81.2	+1.4
10	82	57.4	+14.0	61	85.4	5.6
15	82	57.4	+14.0	62	86.8	+ 7 FT
20	55	38.5	-4.9	60	84	4.2
25	41	28.7	-14.7	57	79.8	0
30	39	27.3	-16.1	55	77	-2.8
35	41	28.7	-14.7	54	75.6	-4.2
40	44	30.8	-12.6	53	74.2	-5.6
45	49	34.3	-9.1	52	72.8	-7
50	51	35.7	-7.7	52	72.8	-7
55	55	38.5	-4.9	52	72.8	-7
60	59	41.3	-2.1	52	72.8	-7
65	60	42.0	-1.4	53	74.2	-5.6
70	62	43.4	0	53	74.2	-5.6
75	63	44.1	+0.7	53	74.2	-5.6
80	64	44.8	+1.4	54	75.6	[illegible]
85	64	44.8	+1.4	54	75.6	[illegible]
90	65	45.5	+2.1	55	77	-2.8
95	65	45.5	+2.1	55	77	-2.8
100	65			56	78.4	-1.4
105	65			56	78.4	-1.4
110	65			56	78.4	[illegible]
115	65			57	79.8	[illegible]
120	65			57	79.8	0
125	65			57	79.8	0
130	65			57	79.8	0

	12-2	ORANGE	STEAM FLOW RATE	TOTAL		
[illegible]	LOW PRESS	HIGH PRESS	TOTAL PRESS	%	LP %	HP %
0	2	53	70	100	100	100
5	16	62	73	111	133	107
7	17	[illegible]	77	110	150	105
10	18	60	77	[illegible]	150	103
15	[illegible]	57	72	103	125	[illegible]
20	7	48	55	77	52	82
25	4	46	50	71	33	79
30	5	46	51	73	42	79
35	6	[illegible]	51	73	33	78
40	7	45	52	74	52	[illegible]
45	8	45	53	76	[illegible]	[illegible]
50	9	46	55	77	55	79
55	10	47	57	81	82	81
60	11	48	57	81	92	83
65	11	48	59	84	92	[illegible]
70	11	50	61	87	92	[illegible]
75	12	51	63	90	[illegible]	88
80	12	[illegible]	64	91	108	86
85	[illegible]	[illegible]	[illegible]	92	[illegible]	[illegible]
90	12	[illegible]	67	96	[illegible]	[illegible]
95	12	[illegible]	[illegible]	97	100	97
100	11	57	67	97	92	[illegible]
105	[illegible]	[illegible]	69	99		[illegible]
110	11	58	[illegible]	[illegible]		[illegible]
115	[illegible]	[illegible]	69	[illegible]		[illegible]
[illegible]			[illegible]	[illegible]		[illegible]
[illegible]			70	[illegible]		
[illegible]	11	[illegible]	71	101		

ICE RAMMING SIMULATION



NOTES 1. DID STEP CHANGES (NOT RAMP) ⟶ MORE SQUARE

2. CONTROLLER USED SIMPLE FEED FORWARD COMMAND
NO LEAD-LAG OR OTHER COMPENSATION
EXCEPT FLOW SPLIT CONTROLLED = 5.0

3. NO TRIM CONTROL USED; WOULD BE TOO SLOW
ANYWAY (AS WE NOW SEE IT) TO HAVE
MUCH EFFECT

4. LOOKS LIKE WE CAN DO IT!

7

TO FILE

RE: Meeting with Foster Wheeler

PLACE: St. Catherines, Ontario

DATE: September 18, 1980

PRESENT: J. Wainwright
T. Thompson
J. Cartier
T. Ruhe
A. Iozsef

TOPIC: Waste Heat Boilers for Combined Cycle Propulsion

---

1. Number of Exhaust Stacks Feeding Each Boiler.

The arguments for one versus two boilers one each gas turbine exhaust stack were discussed. A single boiler being served by two gas turbines offers the following advantages:

- lower capital cost: approximately 80% of two smaller boilers
- lower space requirement: difficult to determine absolute variance in space, but Foster Wheeler recommends 6 feet be provided along the boiler circumference for routine maintenance. The tubing area for the alternative boiler concepts would likely be very similar
- improved control and operation: boiler would not see as many cold to hot cycles

Possible disadvantages of this boiler concept include:

- gas turbulents at exhaust duct juncture: Foster Wheeler related problems of this sort encountered at Fort Saskatchewan, Alberta where a single Frame 7 exhaust into a boiler via ducting which has not been properly designed. This has substantially increased gas turbine back pressure and reduced boiler efficiency.
- little experience with two turbines exhausting, to one boiler: more difficult engineering
- thermal variation of tubes is not expected to be a problem: casing is likely most critical element in terms of non-uniform thermal loading of one large boiler in single gas turbine operation scenario.
- efficiency of boiler is expected to be marginally better with one boiler per turbine due to tubing be optimized for single gas path only

- one boiler for two gas turbines requires engineering of dampers to prevent gas flow rationing through unused turbine
- one boiler per turbine provides greater flexibility in terms of maintaining the boilers
- one boiler per turbine concept requires hot water be circulating during slow times to avoid corrosion of tubing. Circulation of sparsing steam with high pH content is necessary

Conclusion: The considerable time operating with one turbine per shaft suggests the one boiler for two gas turbines may be the preferred concept to avoid circulation of hot water and potential corrosion problems with an unused boiler. Controls are simpler and capital cost is less. Exhaust gas flows must be studied to avoid turbulents and excessive back pressure. This is particularly true when considering supplementary firing of the boiler.

2. Boiler Design

Foster Wheeler recommended the boiler be designed with horizontal tubing and forced or assisted circulation. Horizontal tubing offers less metal (lower cost) and improved operation (less probability of running dry) than vertical tubing. Forced or assisted circulation improves boiler control in maintaining steam flow and pressure. Foster Wheeler also agreed with pinch points, approach temperatures, pressures (single pressure concept) estimated by APP.

3. Supplementary Firing

Foster Wheeler advises that there are no technical difficulties with supplementary firing. Approximately 10% of the boilers they now fabricate include supplementary firing. The restrictions in this regard include limiting gas temperature at boiler entrance to 1400 degrees F., although 1200 degrees F is recommended. At 1200 degrees F, supplementary firing will supply approximately 5000 HP per gas turbine exhaust. No additional oxygen is required to supplement gas turbine exhaust although air supply to boiler without gas turbine operation may be difficult. Supplementary firing will add approximately 20%-25% to the capital cost of the boiler.

4. Thermal Cycling

From the GE LM2500 characteristics, during the MSL ice ramming cycle the boiler will experience exhaust gas variation from 225 degrees F, 165 lb/sec (full power, ambient temp. -25 degrees F) to 640 degrees F, 75 lb/sec (approx. 5-10% load). Foster Wheeler stated this will present little duty on the boiler and the only requirement to handle thermal cycling is to oversize the steam drum to accept water swelling. A typical drum for our steam load might be 48" diameter, 8' to 10' long. To accomodate water swelling the drum may require 72" diameter and possible 100" diameter. The steam drum size can be limited by installing a dump valve to the condenser.

The use of supplementary firing during the ramming cycle to smooth steam production is an alternative to be considered.

5. Boiler Fuel

Natural gas to the boiler supplementary burners should be provided at ambient temperature and 10-15 psig.

6. Boiler Materials

Code requires carbon steel tubing with a dry firing temperature of 800 degrees F, carbon MO tubing with dry firing temperatures of 850 degrees F and chrome MO tubing at higher temperatures. Of the total boiler cost, tubing accounts for approximately 10%. Alloy tubing is approximately 2 to 3 times more expensive than conventional carbon steel piping.

Foster Wheeler recommend carbon steel tubing having 1/16 " additional thickness to accept carbonization that will occur at the higher gas temperature during dry firing of the boiler. This will add approximately 5% to the total boiler cost.

7. Auxilliary Boilers

Foster Wheeler suggested that rather than consider supplementary firing, auxilliary boilers be installed to provide additional power when required. An auxilliary boiler having steam production capability of 50,000 lb/hr would cost approximately $225,000.

8. Capital Cost Inflation

The waste heat boiler prices provided by Foster Wheeler in September 1979 should be inflated by 12% to make them valid to September 1980.

56
79-09

709

INTERNAL CORRESPONDENCE

DATE 80/10/17
YEAR MONTH DAY

To FILE

FROM JOHN WAINWRIGHT

DEPARTMENT OR FIELD OFFICE

COST CENTRE OR AFE No

FILE No.

SUBJECT RE: Combined Cycle Propulsion

During the week of October 6 - 10, Ted Thompson and I visited Solar Turbines International and The Garrett Corporation to learn their progress with the "Rankin Cycle Energy Recovery" (RACER) development programme being conducted for the U.S. Navy. Both firms had recently completed the initial or design phase of the programme and submitted their reports to the Navy. The Navy will now evaluate these two reports, and a third submitted by General Electric, and will, within six months, award a contract to one of these firms for development of the hardware. The following paragraphs describe the seminars held with Solar and Garrett.

Monday, October 6
Solar Turbines International Offices
San Diego, California

Present: Norm McIntyre . . . Solar
Geoff Clough . . . Solar
Ted Thompson
John Wainwright

The morning session consisted of a Solar presentation describing their combined cycle design (refer attached presentation slides). The Solar system is a "once-through" boiler which does not include a steam drum. Control of the steam system is slightly more complicated than a conventional system but this is off-set by the advantage of fewer components.

The Solar combined cycle design is predicated upon achieving maximum reliability. As a result, boiler tubing is Incoloy 800, and the condenser sheeting is titanium. The system has been designed to minimize steam leakage and thus the water treatment necessary. The steam loss of the system is estimated at 5 gallons per day and most of this will occur through the turbine seal. The system does not include water pretreatment; rather distilled water will be used as make-up and will be de-ionized only.

The system has been designed to 100° F pinch point, 80° F - 100° F approach temperature, and a minimum gas exhaust temperature of 300° F.

The system may be dry fired, will produce steam in approximately 4 minutes and will be at full operation in 20 minutes.

Solar has developed reliability figures for their steam system and these are shown below. The control system is the weakest link in the system.

	MTBF (Hrs)
Controls and Instrumentation	30,121
Condenser System	60,241
Feedwater System	39,841
Waste Heat Boiler	90,090
Hot Valve Module	192,308
Steam Turbine	188,679
Total Mean Time Between Failure	10,363

The system mean time to repair is estimated at 2 hours.

The boiler is expected to add 91/2" water pressure loss on the outlet duct.

The afternoon session consisted of a tour of the proto-type combined cycle plant developed by Solar at their plant on the outskirts of San Diego. This unit has been placed on the exhaust of the Centaur gas turbine. Solar has recently signed three contracts for sale of these Centaur combined cycle units.

Additional discussion concerned the means of supplying ship service steam and ballast heating requirements from the closed loop combined cycle. Although many alternatives were discussed, none seemed particularly appropriate for our icebreakers.

Tuesday, October 7
Solar Offices - San Diego
same attendees as previous meeting

This meeting discussed in detail the simulation of an LM - 2500 combined cycle plant undertaking icebreaking operations. These simulations were conducted by Solar Turbines International on behalf of the APP. The simulation indicated the "once-through" boiler would have absolutely no difficulty in undertaking the ramming cycle (with fixed pitch propellers). The boiler will see little thermal stress, excess steam may be dumped directly to the condenser and the "dry out" zone will move slightly but not sufficiently to cause any tubing stress. In summary, the icebreaking operations will not affect the useful life or performance of the boiler according to the simulation. These results will be described in much greater detail in a separate memo.

Wednesday, October 8
Garrett Corporation Offices - Los Angeles

Robert Chapman	Garrett
Robin MacKay	Garrett
Frank Halpen	Garrett
Bill Miller	Garrett
	~~Garrett~~

Ted Thompson
John Wainwright

The meeting discussed the Garrett Racer programme and regenerator development. Other items of discussion included Garrett's research into superconducting motors, air foil bearings, small gas turbines for ship service power, and the means of vapourizing LNG.

The remainder of the day involved a tour of Garrett research facilities and production plants.

Thursday, October 9
Garrett Corporation Offices - Phoenix
Robert Chapman
Robin MacKay
Peter Donnelly
Steve Trimble
Ted Mock
Ted Thompson
John Wainwright

The meeting reviewed development of the Garrett RACER for the U.S. Navy. Garrett has developed a conventional boiler design for their combined cycle system. Their boiler design includes the steam drum between evaporator and superheater.

The Garrett system may be dry fired and also has a theoretical mean time between failure of over 10,000 hours. This latter figure is accomplished in part by a closed steam system that will only require approximately 2 gallons of make-up feedwater per day.

The attached brochure describes their system in greater detail.

Garrett then provided their reply to many questions concerning combined cycle propulsion that I had asked many months previously. Their reply is described below:

i) Number of boilers - Garrett recommends one boiler per gas turbine exhaust. Upon further examination, they have not great rationale for this statement other than flexibility.

ii) Supplementary Firing - Garrett do not recommend supplementary firing. They feel burners cannot be located in a position where they would be effective and reliable.

iii) Dry Firing - Garrett has designed their boiler to accept dry firing and see no difficulty in operating in this manner.

iv) Waste Heat Boiler Control - Little control problem as outlined in Garrett's brochure

v) Icebreaking Operations - The Garrett evaluation concluded the ice ramming cycle and other icebreaking operations would have vertually no affect on the boiler.

vi) Reliability - Garrett estimates the following for the combined cycle system

MTBF . . . . over 10,000 Hrs
MTTR . . . . 2 Hrs
Availability 99.97%
Expected Routine Maintenance. . 5.9 manhours/week

DESCRIPTION OF RACER DEVELOPMENT PROGRAMME

[illegible]

Urgent National Need

- Save Natural Resources
- Protect the National Economy
- Provide Basic Economic Strength for the [illegible]

REC'D FROM SOLAR TURBINES
INTERNATIONAL. 10/1980

RACER

Uses gas turbine exhaust heat usually wasted to generate steam for additional power.

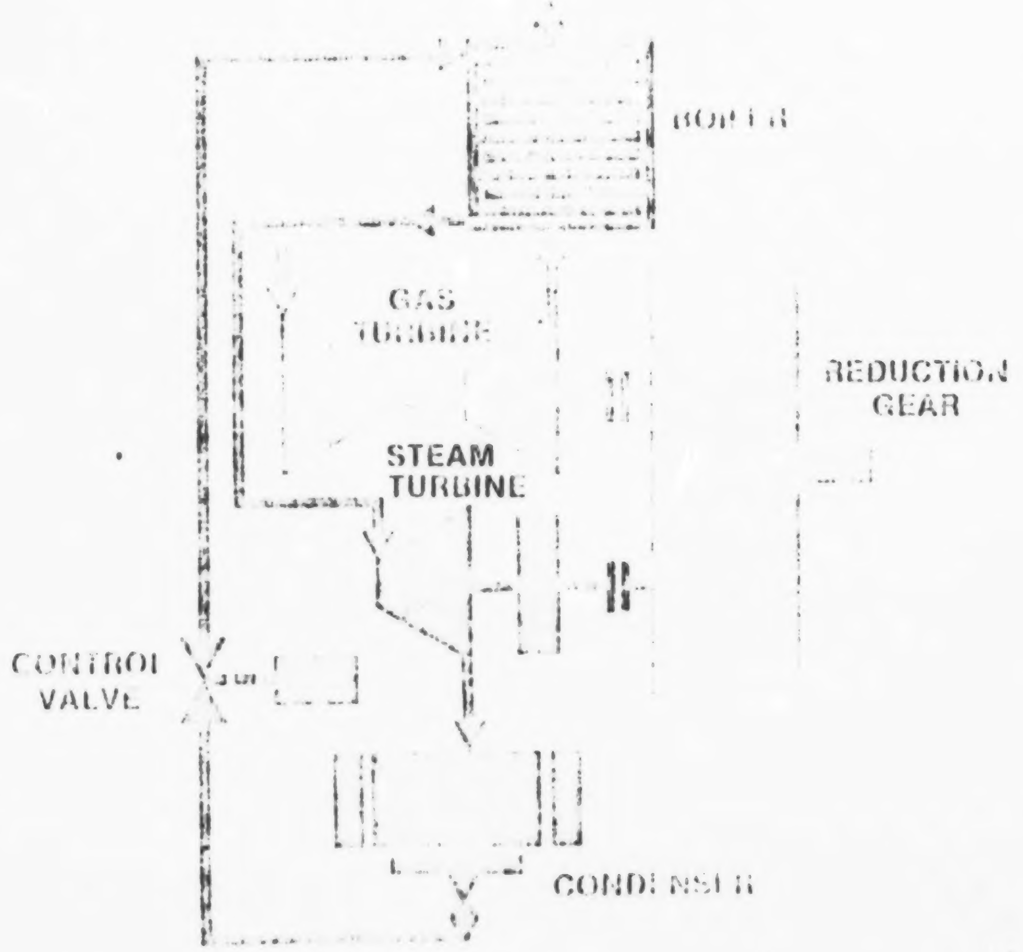
U.S. NAVY RACER PROGRAM

Purpose:

To develop a high efficiency cruise propulsion system for non-nuclear surface combatants with LM2500 main propulsion gas turbines

Visual Products Divi

Combined Cycle Power Plant


BOILER
GAS TURBINE
REDUCTION GEAR
STEAM TURBINE
CONTROL VALVE
CONDENSER

[illegible]

HACER Requirements

Same as Solar Commercial Combined Cycle Plants

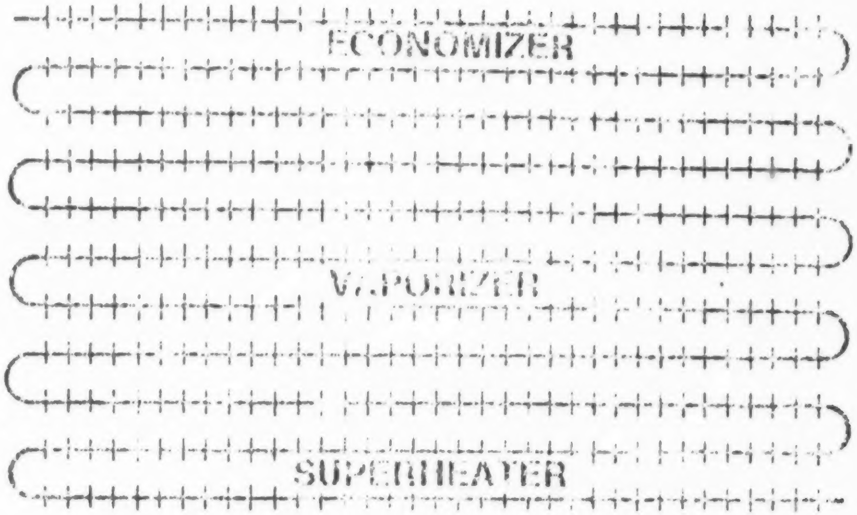
- Low Manpower Requirements
- High Reliability
- Easily Maintained
- Low Weight
- High Performance
- Simple

PRIMARY APPROACH

- Simplicity
- Modularity

Once-Through Boiler

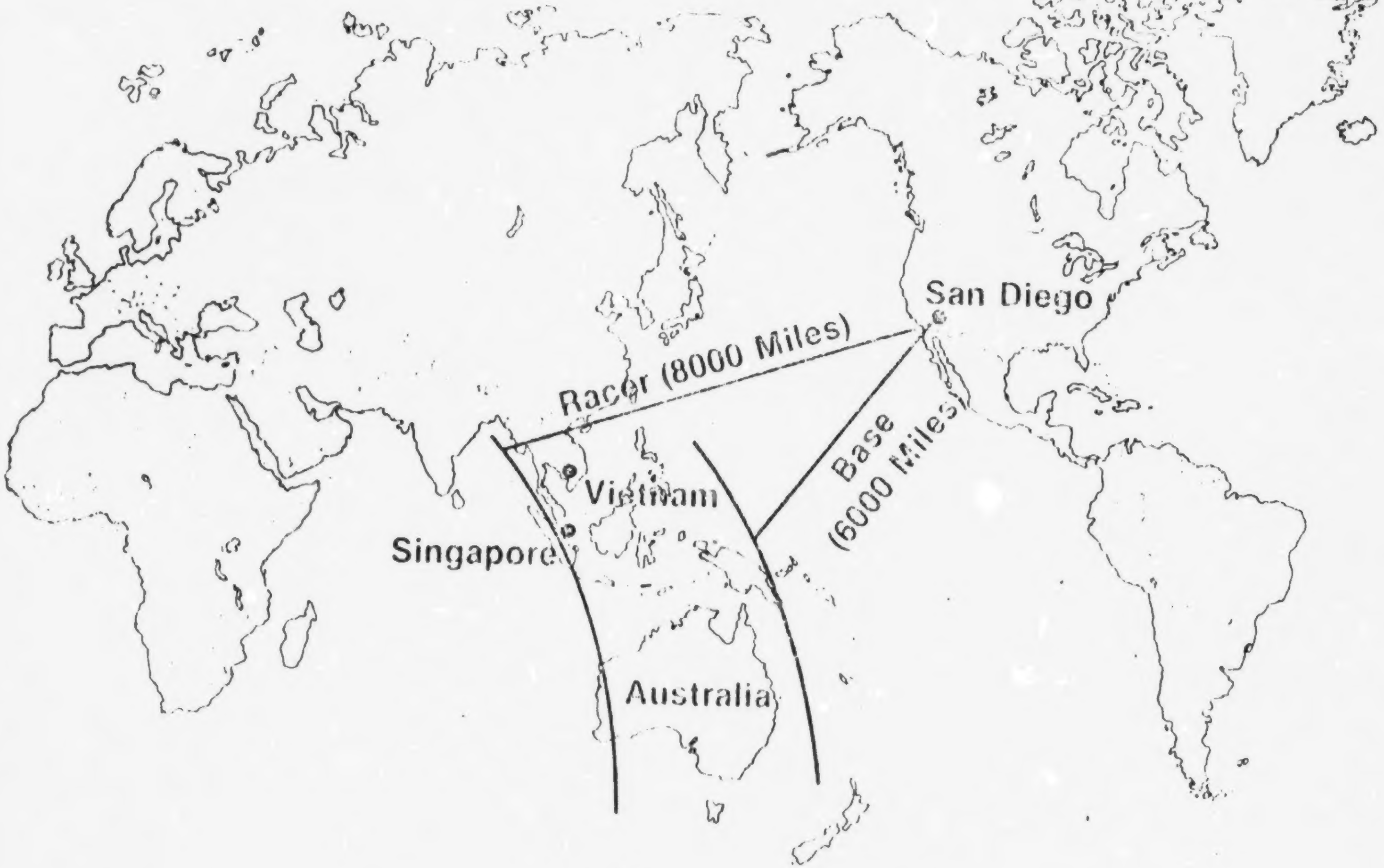
- Maximum Performance Per Unit Weight
- Maximum Performance Per Unit Volume
- Low Water Inventory
- Simple Controls
- Modular Concept


ECONOMIZER
VAPORIZER
SUPERHEATER

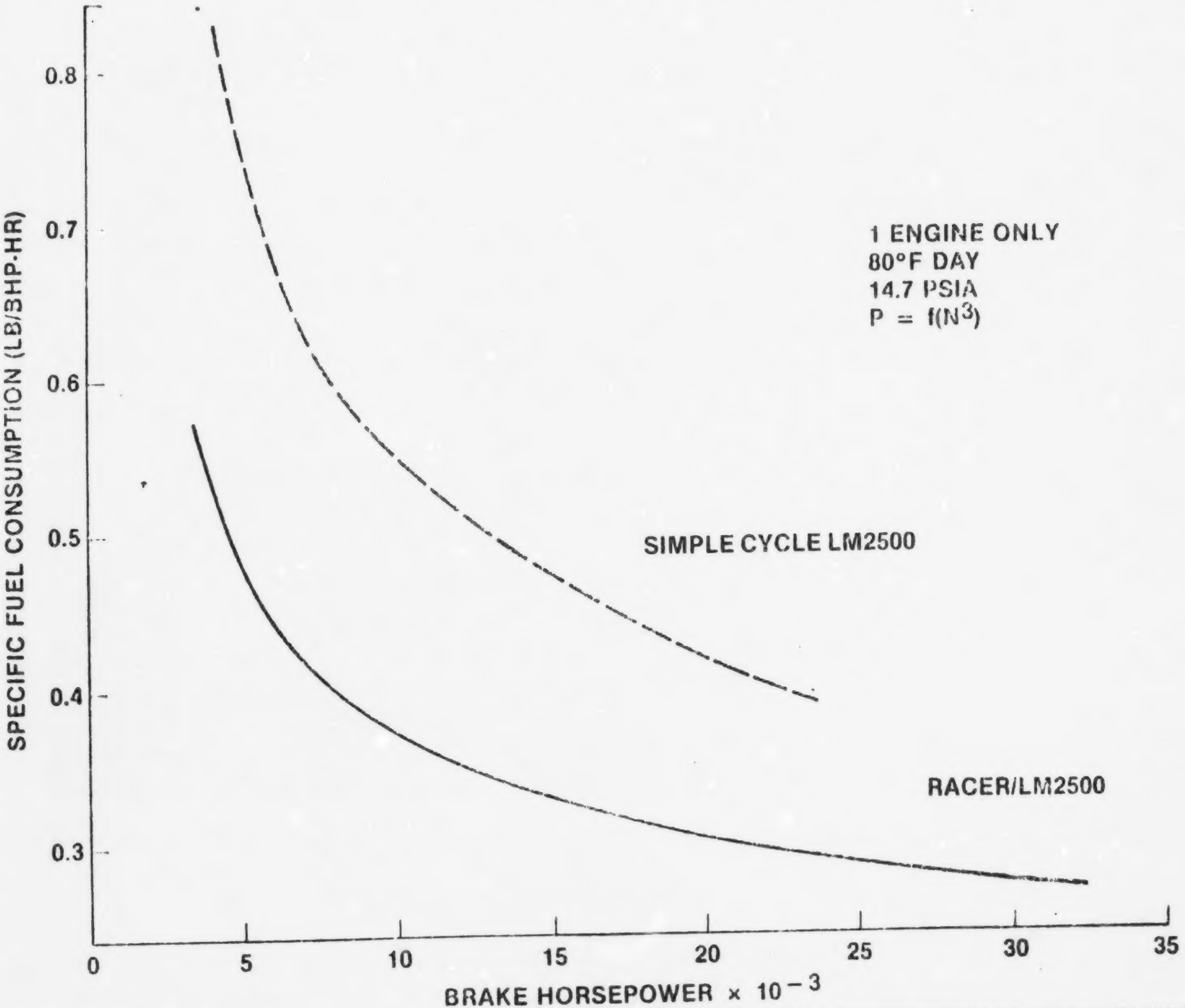
Racer System

- Components Installed Through Inlet Stack
- Very Simple Piping System
- Fully-Automatic Single-Valve Control System
- Clutched Steam Turbine for Improved Machinery Availability
  - Reliability
  - Maintainability
- Unmanned Operation

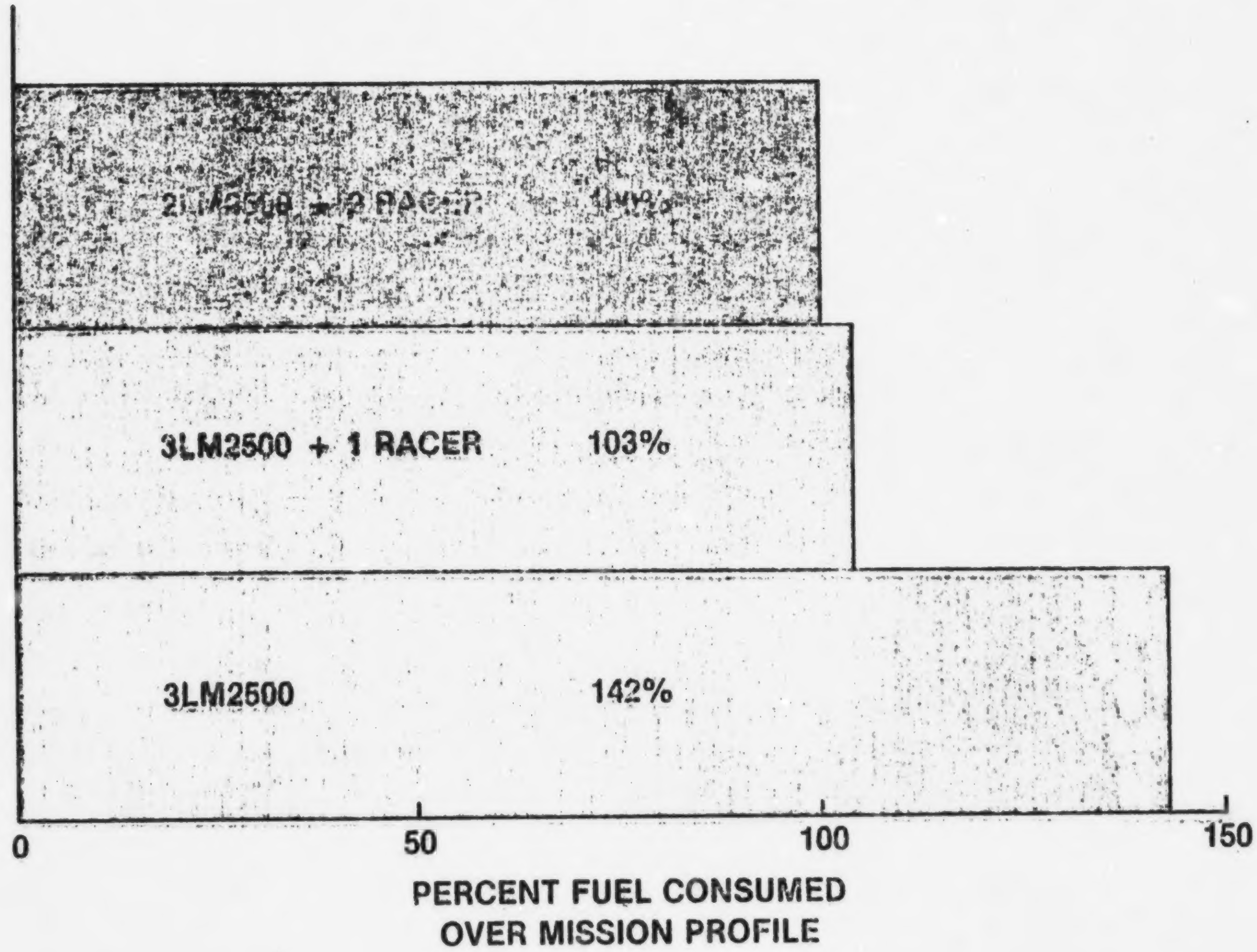
RACER Improves Destroyer Range

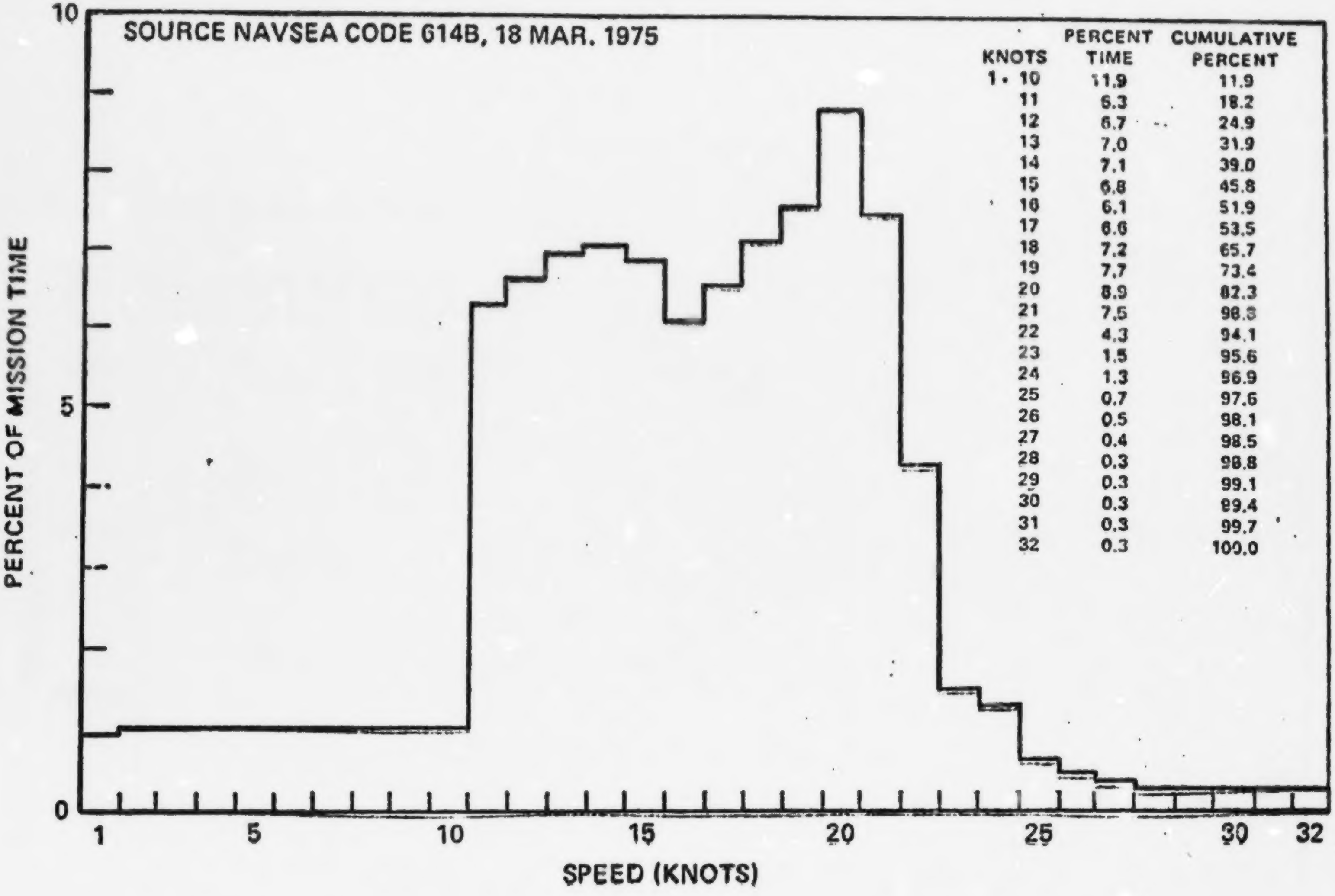

San Diego
Racer (8000 Miles)
Base
(6000 Miles)
Vietnam
Singapore
Australia

COMPARATIVE FUEL RATES



COMPARATIVE DDGX ANNUAL FUEL USAGE





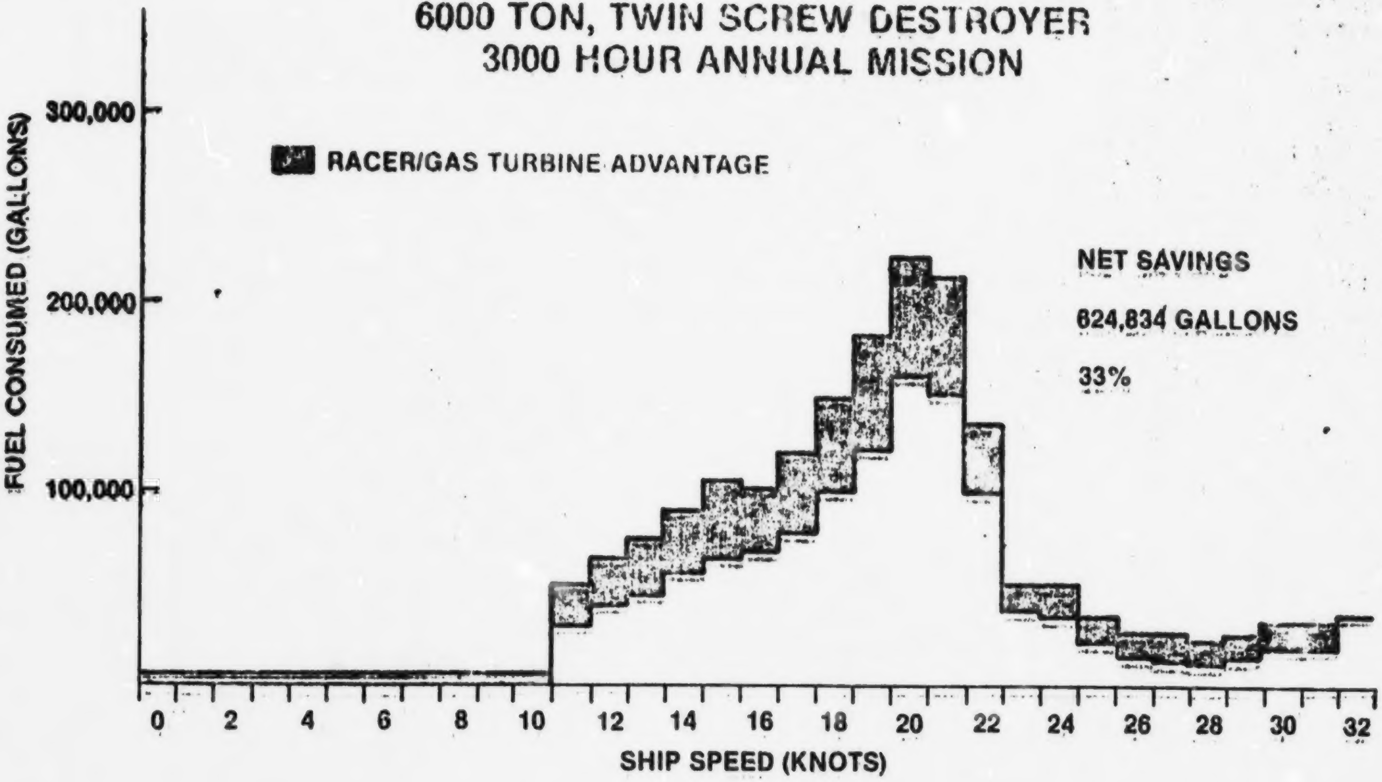
SOURCE NAVSEA CODE 614B, 18 MAR. 1975

KNOTS	PERCENT TIME	CUMULATIVE PERCENT
1 - 10	11.9	11.9
11	6.3	18.2
12	6.7	24.9
13	7.0	31.9
14	7.1	39.0
15	6.8	45.8
16	6.1	51.9
17	6.6	53.5
18	7.2	65.7
19	7.7	73.4
20	8.9	82.3
21	7.5	98.3
22	4.3	94.1
23	1.5	95.6
24	1.3	96.9
25	0.7	97.6
26	0.5	98.1
27	0.4	98.5
28	0.3	98.8
29	0.3	99.1
30	0.3	89.4
31	0.3	99.7
32	0.3	100.0

FIGURE 13 STANDARD NAVSEC DESTROYER PROFILE

COMPARATIVE FUEL USAGE

RACER VS GAS TURBINE
6000 TON, TWIN SCREW DESTROYER
3000 HOUR ANNUAL MISSION



NET SAVINGS

624,834 GALLONS

33%

SOLAR'S
COMBINED CYCLE TEST FACILITY
PERFORMANCE

Predicted Thermal Efficiency	37%
Actual Thermal Efficiency	40%

Solar's Combined Cycle Test Facility

1969 — Started Development of
Once-through Boiler

1973 — Committed to Combined
Cycle System

1976 — Completed Stability
Tests on Boiler

1978 — Started Construction of
Combined Cycle Test Facility

1980 — Combined Cycle Test
Facility Operational

SOLAR TURBINES INTERNATIONAL

Technological Contributions

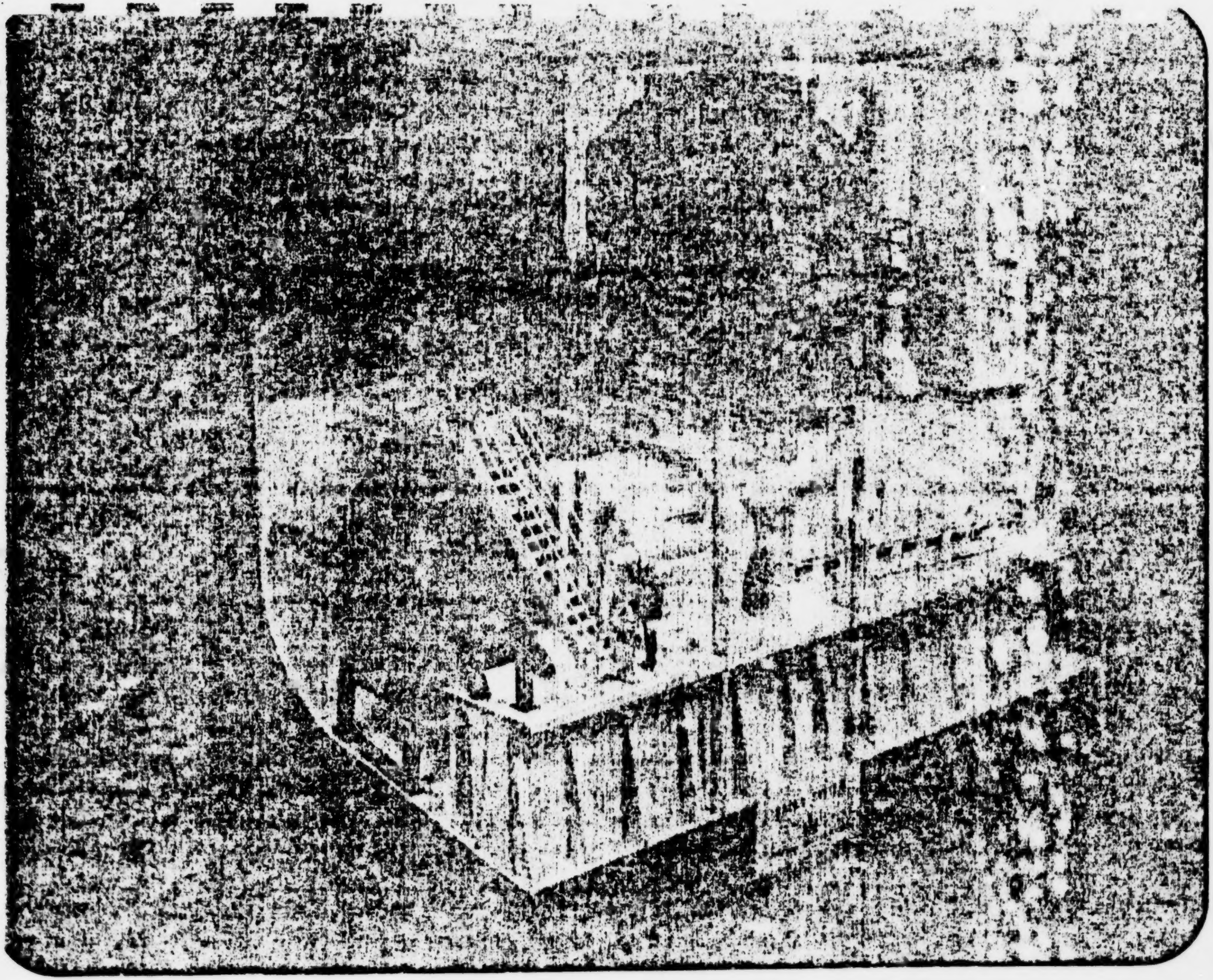
30 Years of Solving Problems in Use of Gas Turbines at Sea

ransparency
nounting frame

SUMMARY

Solar's RACER

- Proven Concept
- 33% Fuel Savings at Cruise
- Most Efficient Plant (Cruise and Full Power)
- High Reliability
- Low Maintenance
- Minimum Weight Impact
- Ideal Destroyer Propulsion Plant

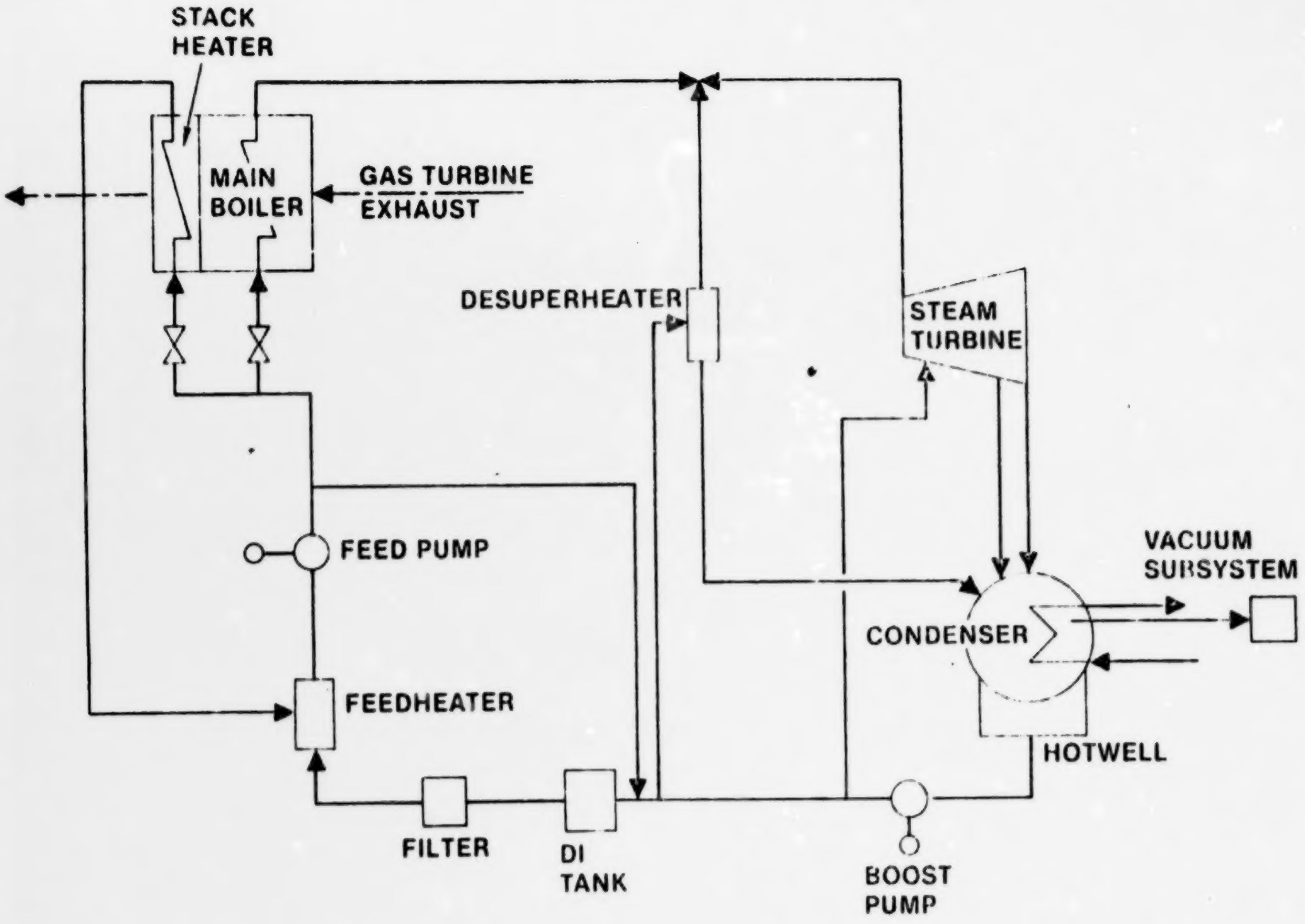


K+E

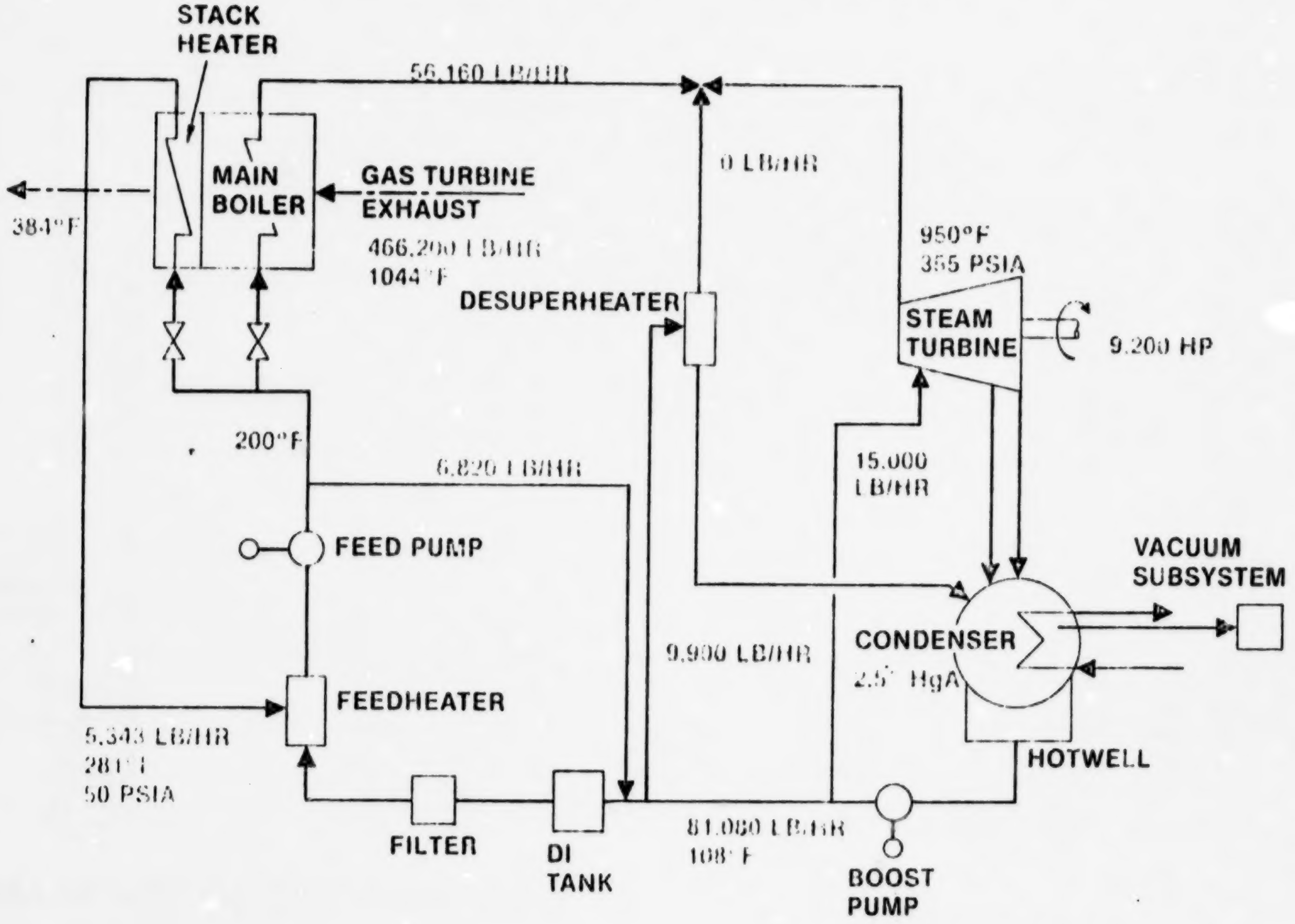
VISUAL
EDUCATION
PRODUCTS

GEMINI
Transparency Mounts

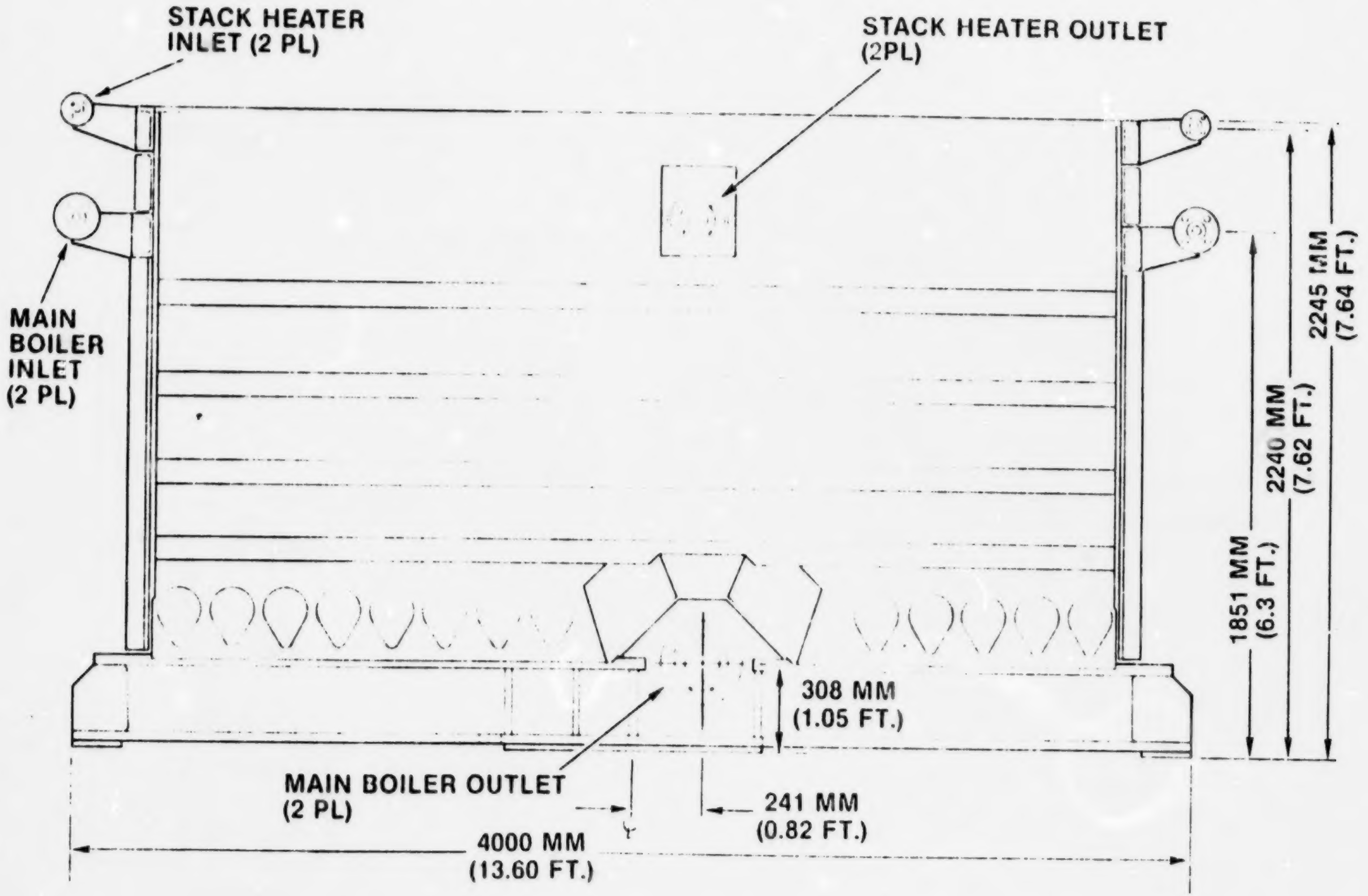
RACER SYSTEM SCHEMATIC


STACK HEATER
MAIN BOILER
GAS TURBINE EXHAUST
DESUPERHEATER
STEAM TURBINE
FEED PUMP
VACUUM SUBSYSTEM
CONDENSER
FEEDHEATER
HOTWELL
FILTER
DI TANK
BOOST PUMP

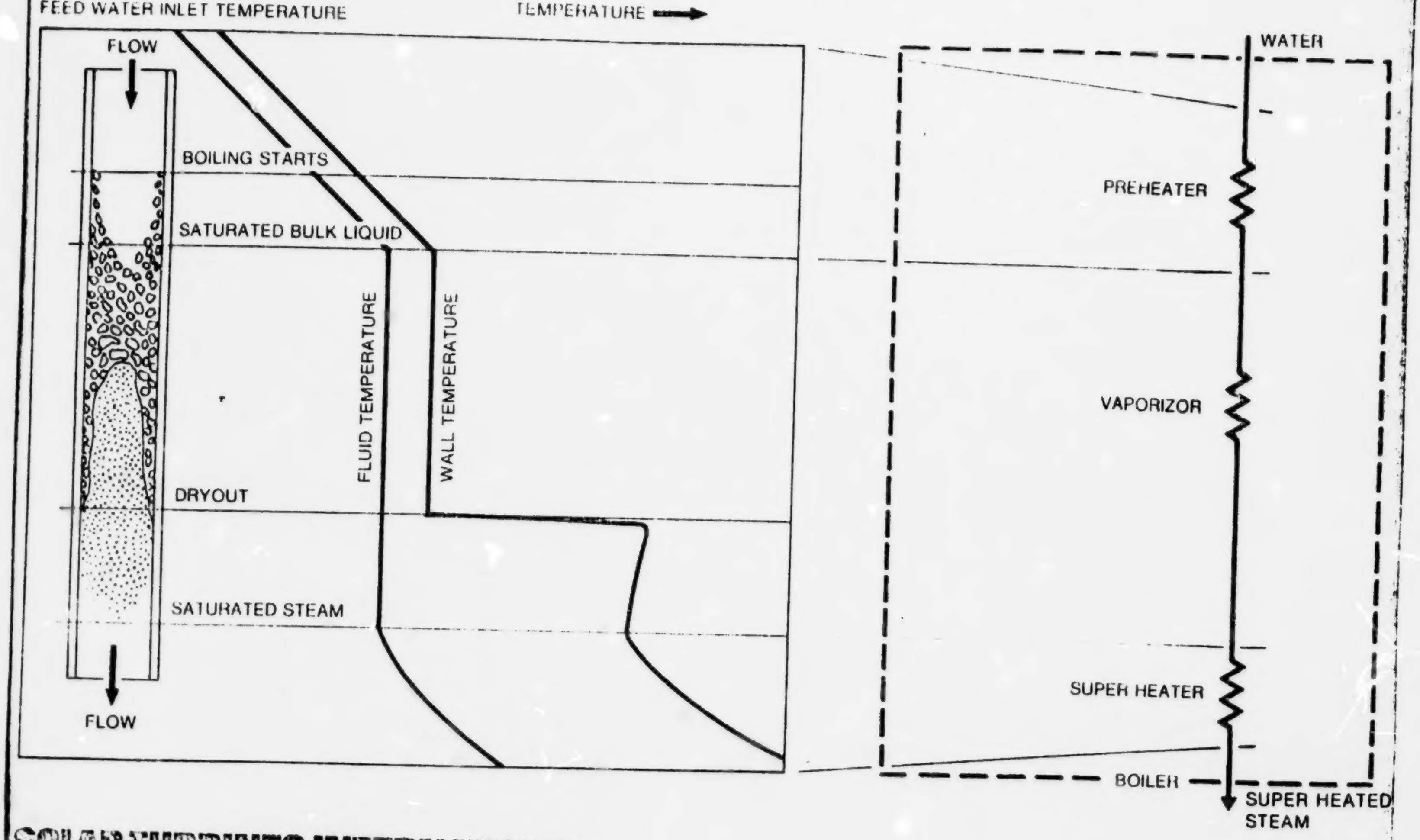
RACER CYCLE PARAMETERS
(HOT DAY @ 32 KTS)


STACK HEATER
56,160 LB/HR
MAIN BOILER
GAS TURBINE EXHAUST
466,200 LB/HR
1044°F
384°F
0 LB/HR
DESUPERHEATER
950°F
355 PSIA
STEAM TURBINE
9,200 HP
200°F
6,820 LB/HR
15,000 LB/HR
FEED PUMP
VACUUM SUBSYSTEM
CONDENSER
2.5" HgA
9,900 LB/HR
FEEDHEATER
5,343 LB/HR
281°F
50 PSIA
HOTWELL
FILTER
DI TANK
81,080 LB/HR
108°F
BOOST PUMP

BOILER OUTLINE

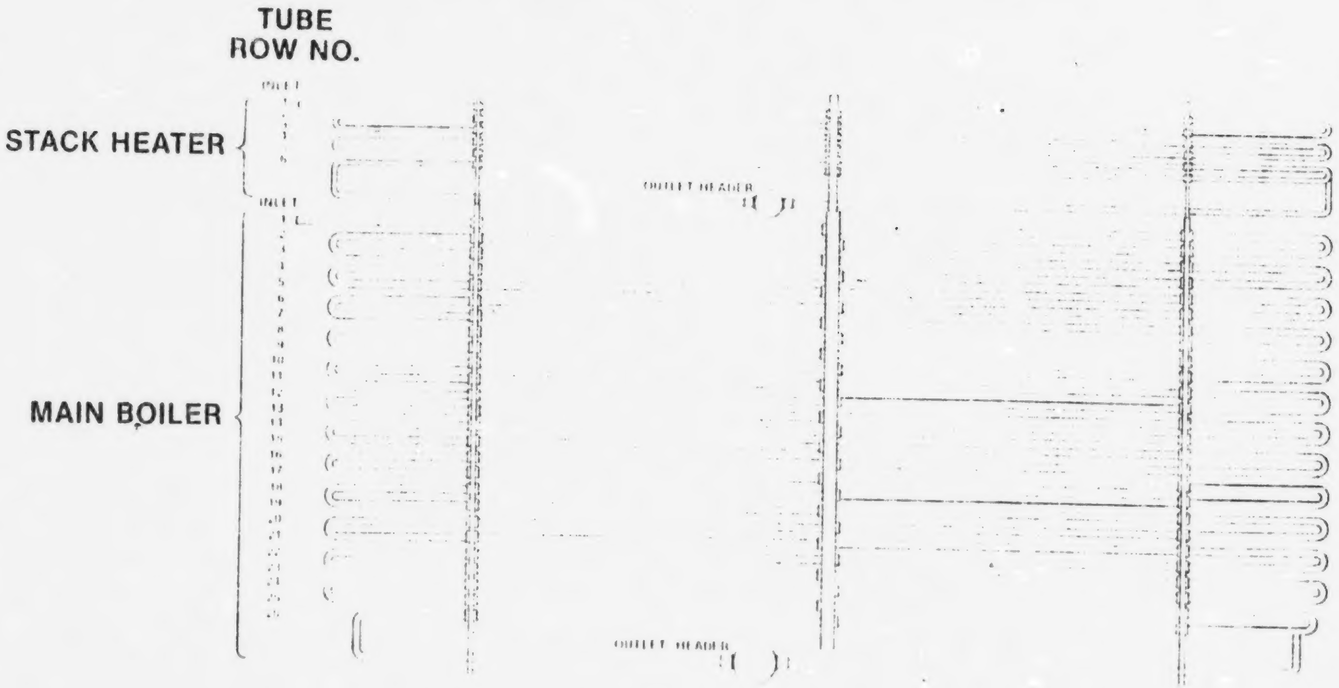

STACK HEATER
INLET (2 PL)
STACK HEATER OUTLET
(2PL)
MAIN
BOILER
INLET
(2 PL)
2245 MM
(7.64 FT.)
2240 MM
(7.62 FT.)
1851 MM
(6.3 FT.)
308 MM
(1.05 FT.)
MAIN BOILER OUTLET
(2 PL)
241 MM
(0.82 FT.)
4000 MM
(13.60 FT.)

Once-Through Boiler Concept

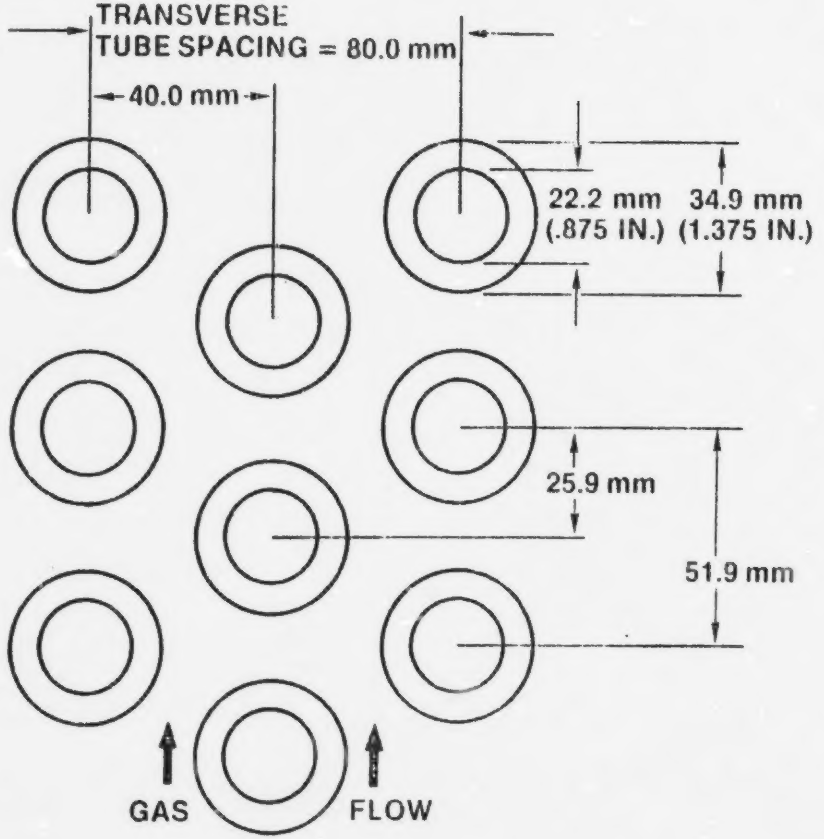

FEED WATER INLET TEMPERATURE
TEMPERATURE
FLOW
BOILING STARTS
SATURATED BULK LIQUID
FLUID TEMPERATURE
WALL TEMPERATURE
DRYOUT
SATURATED STEAM
FLOW
WATER
PREHEATER
VAPORIZOR
SUPER HEATER
BOILER
SUPER HEATED STEAM

SOLAR TURBINES INTERNATIONAL
An Operating Group of International Harvester

RACER BOILER CIRCUIT


TUBE
ROW NO.
INLET
STACK HEATER
INLET
MAIN BOILER
OUTLET HEADER
OUTLET HEADER

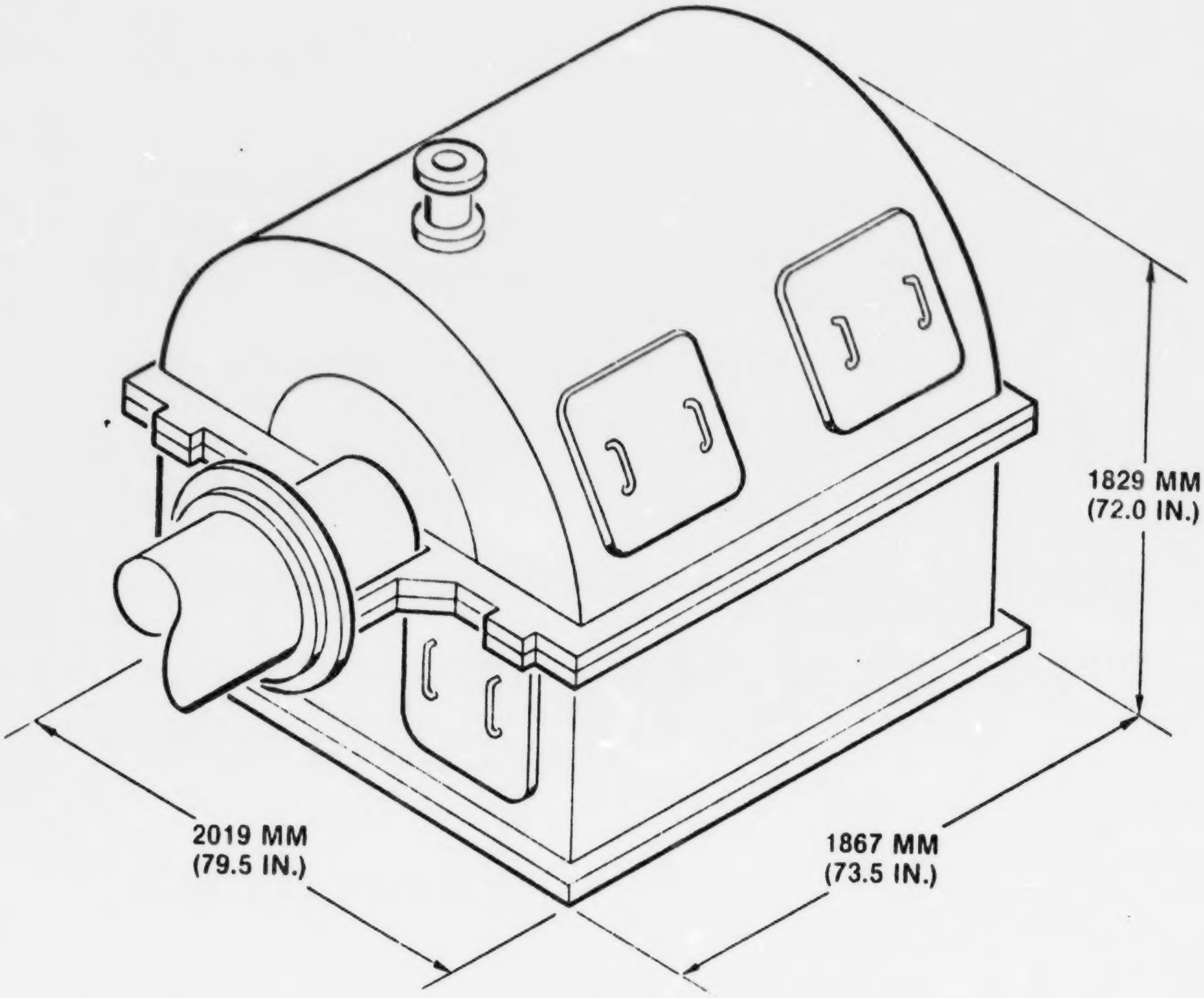
MAIN BOILER TUBE MATRIX CONFIGURATION


TRANSVERSE
TUBE SPACING = 80.0 mm
40.0 mm
22.2 mm
(.875 IN.)
34.9 mm
(1.375 IN.)
25.9 mm
51.9 mm
GAS
FLOW

BOILER DESIGN PARAMETERS

BOILER GEOMETRIES	MAIN BOILER	STACK HEATER
NO. TUBES PER ROW	76	103
TOTAL NO. ROWS (GAS PASSES)	26	6
TUBE DIAMETER, IN.	0.875	0.625
TUBE WALL THICKNESS, IN.	0.049	0.042
FIN DIAMETER, IN.	1.375	0.9375
FIN THICKNESS, IN.	0.016	0.016
PREHEATER — NO. ROWS	8	1
PREHEATER — FINS PER IN.	5.3	5.3
VAPORIZER — NO. ROWS	11	4
FPI	7.6	5.3
SUPERHEATER — NO. ROWS	7	1
FPI	10.1	5.3
ESTIMATED FULL LOAD PRESSURE DROP, IN. H_20	9.03	0.53

RACER STEAM TURBINE


1829 MM
(72.0 IN.)
2019 MM
(79.5 IN.)
1867 MM
(73.5 IN.)

STEAM TURBINE CHARACTERISTICS

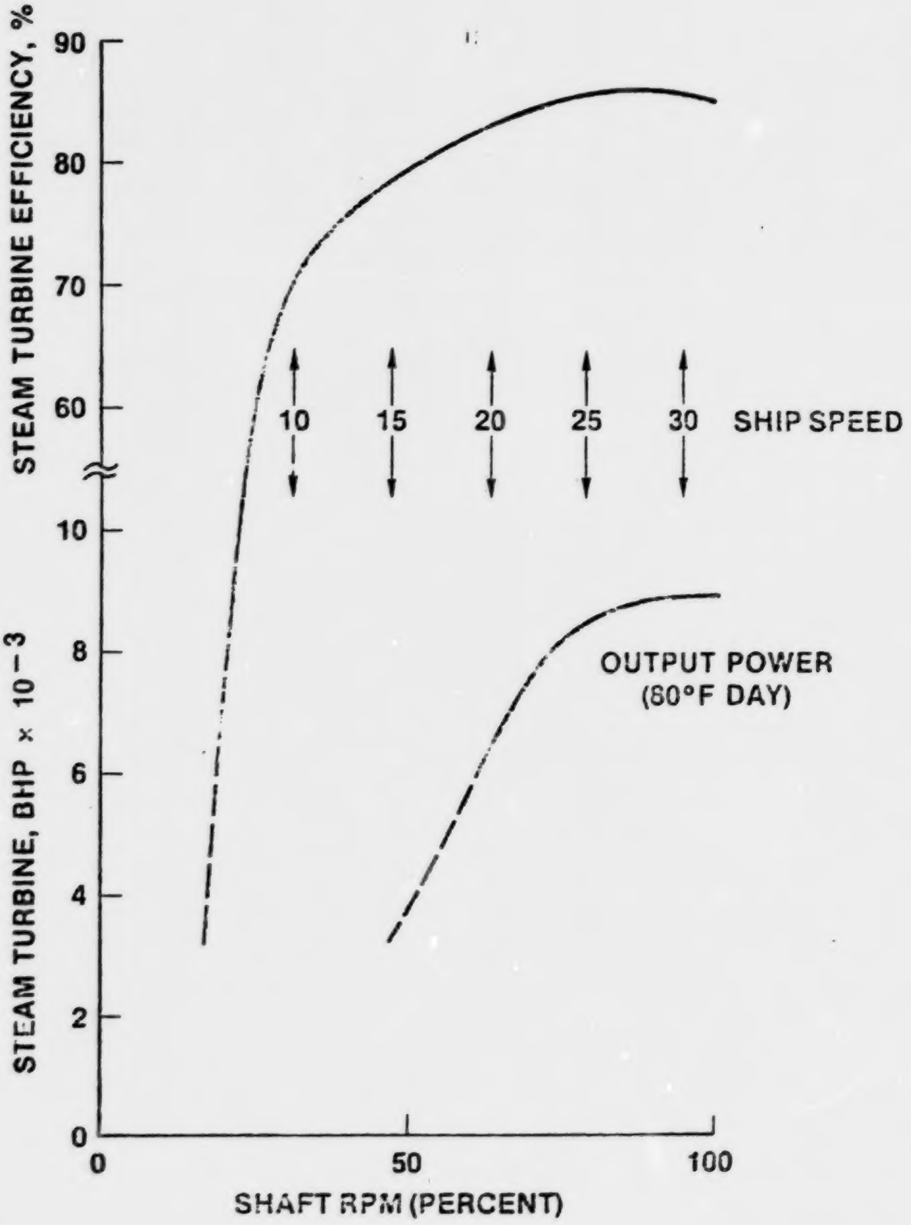
- 11 STAGE AXIAL TURBINE
- 1ST TWO STAGES — PARTIAL ARC ADMISSION
- LAST TWO BLADE STAGES -- TITANIUM
- 10,000 RPM FULL POWER SPEED
- DOUBLE CASING FOR SHOCK AND THERMAL INTEGRITY
- VIBRATION, THRUST, SPEED AND TEMPERATURE SENSORS FOR AUTOMATIC CONTROL AND HEALTH MONITORING

STEAM TURBINE CHARACTERISTICS

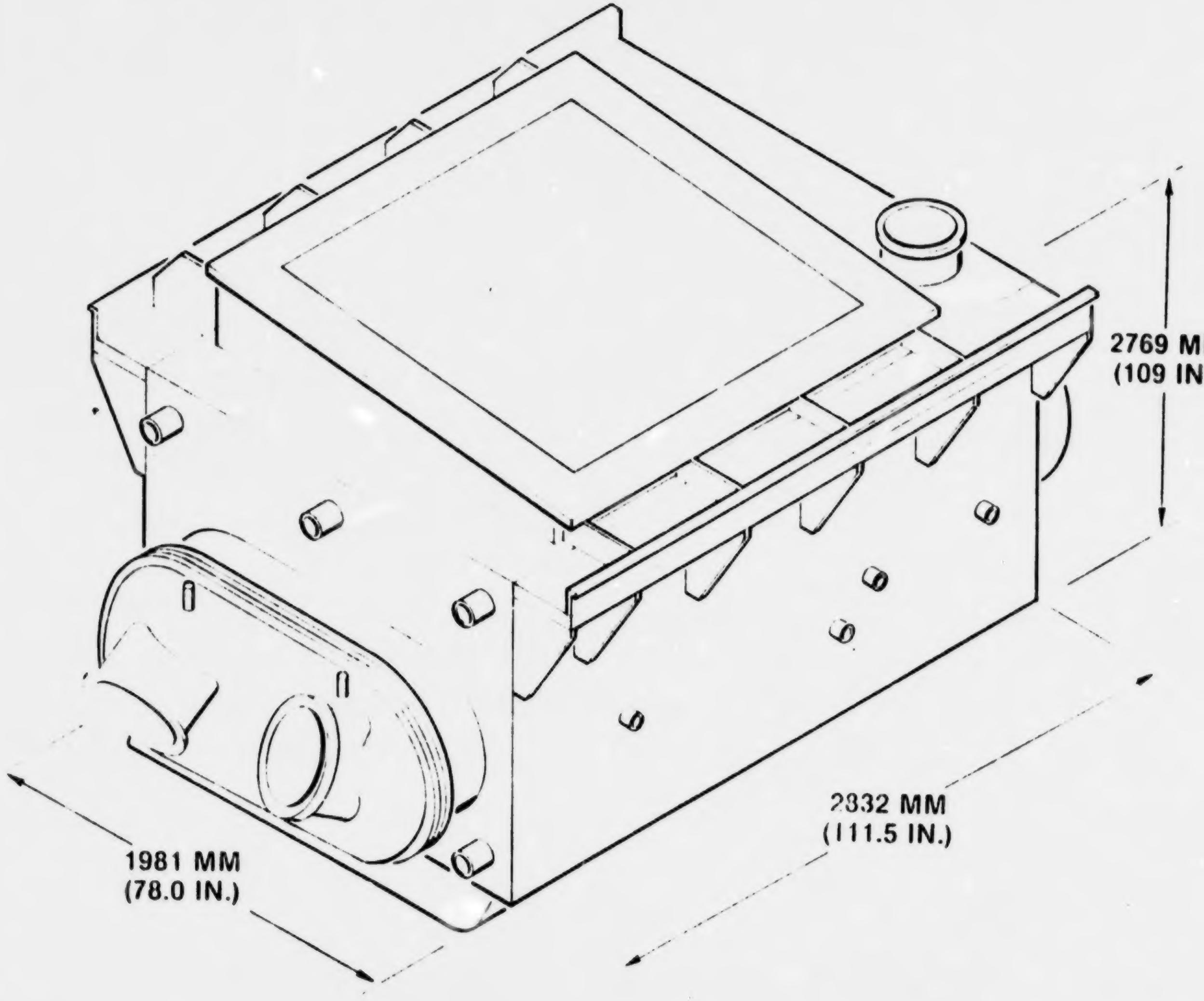
- BORESCOPE PORTS FOR INTERNAL INSPECTION
- WATER FACE SEALS MINIMIZE LEAKAGE RATE
- WATER LUBRICATED BEARING AT L.P. END ELIMINATES L.O. CONTAMINATION POTENTIAL
- 85% PEAK TURBINE EFFICIENCY (TOTAL-TO-STATIC)
- TURBINE EFFICIENCY OVERPEAKED FOR MAXIMUM FUEL SAVINGS OVER MISSION PROFILE

STEAM TURBINE EFFICIENCY AND OUTPUT POWER

(AS A FUNCTION OF PROPELLER RPM)



RACER CONDENSER


2769 MM
(109 IN.)
2332 MM
(111.5 IN.)
1981 MM
(78.0 IN.)

Visual Products Division 3
St. Paul, MN 55101 Made in U.

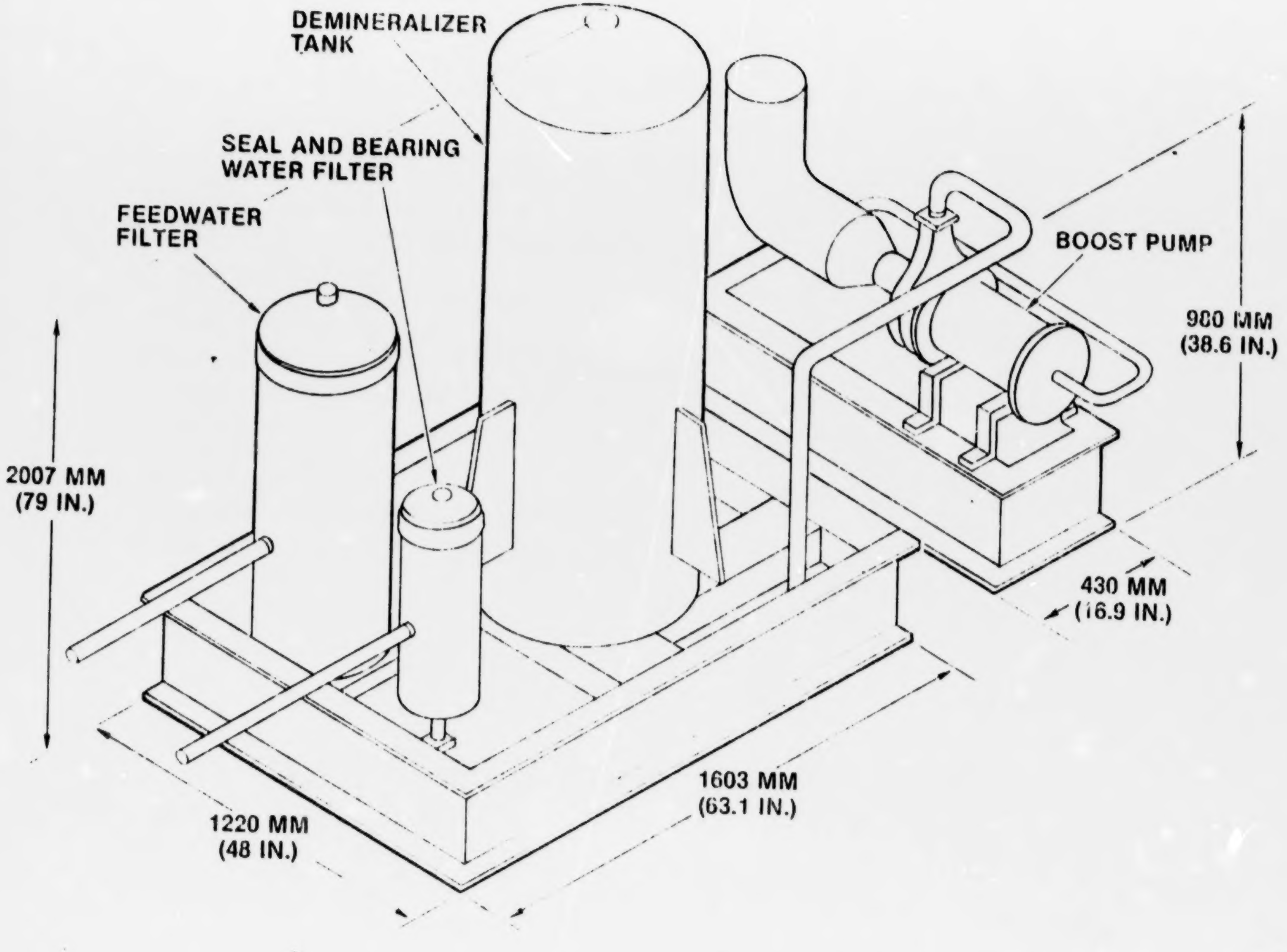
CONDENSER PHYSICAL DATA

HOTWELL CAPACITY (OPERATING)	40 CU. FT.
TOTAL CONDENSER SURFACE	2,950 SQ. FT.
NUMBER OF WATER PASSES	TWO (2)
TUBE MATERIAL	TITANIUM
TUBE SIZE	5/8" - 22 BWG
EFFECTIVE LENGTH OF TUBES	10'0"
TOTAL NUMBER OF TUBES	1,804
OUTER TUBE SHEET MATERIAL	TI CLAD STN. STL.
SHELL MATERIAL	STN. STL.
OPERATING WEIGHT	21,100 LB
DRY WEIGHT OF CONDENSER	15,400 LB

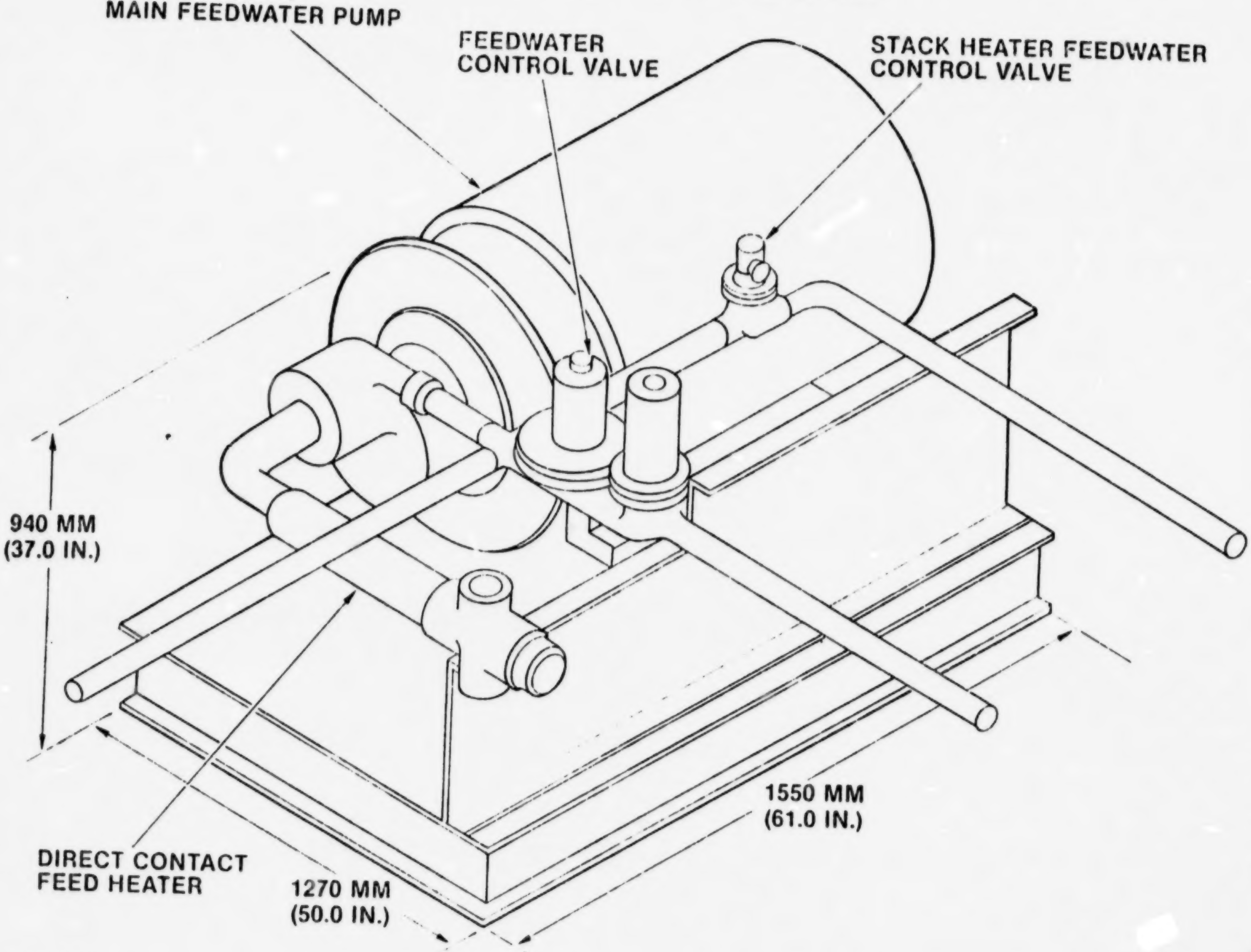
CONDENSER PERFORMANCE DATA

	FULL POWER	20 KTS
TOTAL HEAT LOAD TO CIRC. WATER, BTU/HR	54,600,000	36,900,000
ABSOLUTE PRESSURE AT COND. INLET, IN. HgA	2.5	1.85
CIRC. WATER FLOW, GPM	8,560	8,560
VELOCITY OF WATER IN TUBES, FPS	12	12
TEMPERATURE RISE, °F	12.7	8.6
TEMPERATURE OF STEAM BEING CONDENSED, °F	108.7	98.5
HEAT TRANSFER RATE, BTU/HR/SQ FT./°F	691	691
CLEANLINESS FACTOR, %	90	90
CIRC. WATER PRESSURE DROP, FT. H_2O	34.6	34.6

CONDENSATE MODULE AND PUMP


DEMINERALIZER TANK
SEAL AND BEARING WATER FILTER
FEEDWATER FILTER
BOOST PUMP
900 MM (38.6 IN.)
2007 MM (79 IN.)
430 MM (16.9 IN.)
1603 MM (63.1 IN.)
1220 MM (48 IN.)

FEEDWATER MODULE


MAIN FEEDWATER PUMP
FEEDWATER CONTROL VALVE
STACK HEATER FEEDWATER CONTROL VALVE
940 MM (37.0 IN.)
DIRECT CONTACT FEED HEATER
1270 MM (50.0 IN.)
1550 MM (61.0 IN.)

RACER WATER TREATMENT

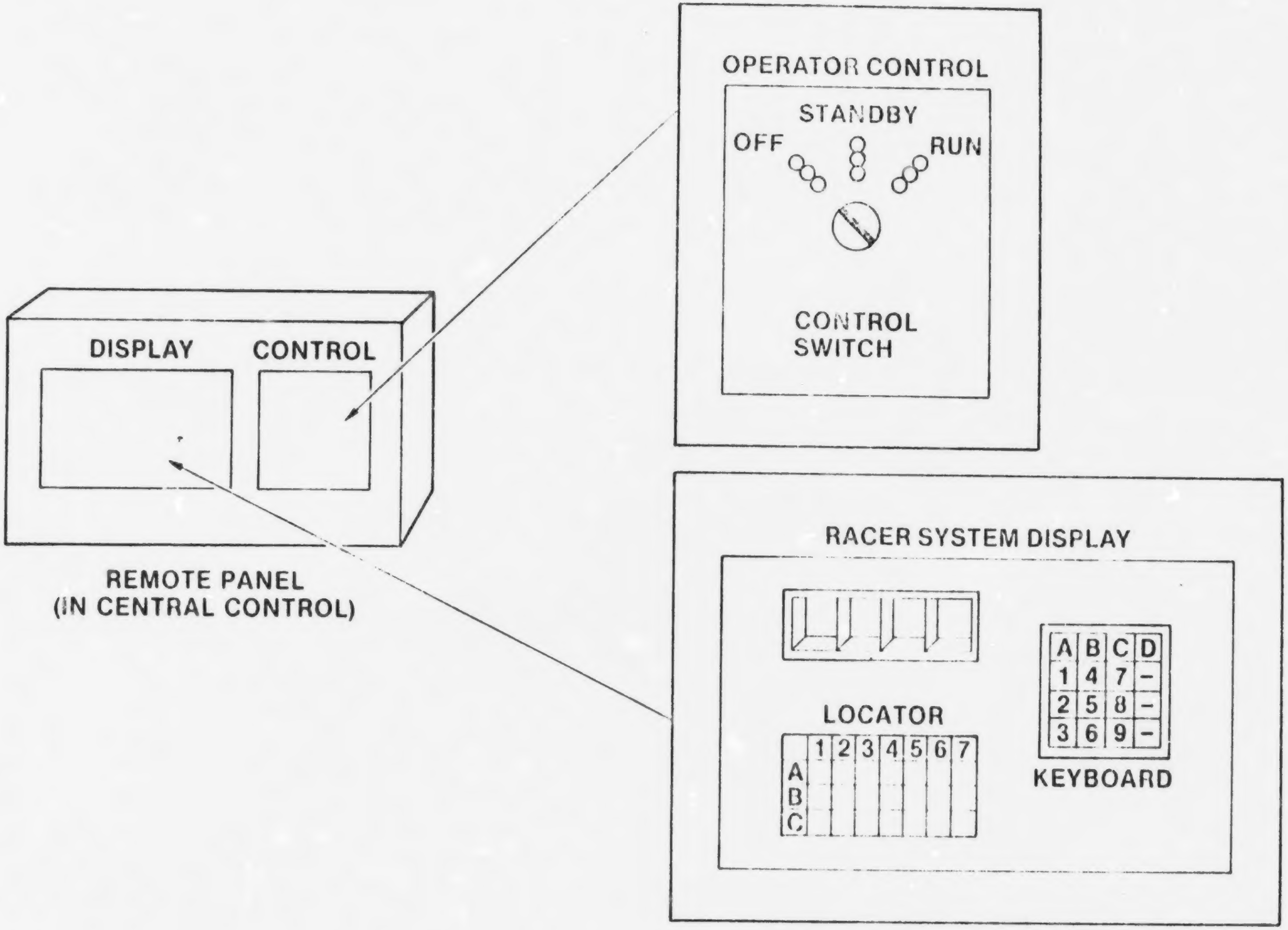
- ANNUAL ION EXCHANGER RECHARGE
- MONTHLY CONDUCTIVITY CELL INSPECTION
- DAILY CONDUCTIVITY READING
- NO CHEMICAL ADDITIONS
- NO SAMPLING
- NO REAGENTS OR STANDARDS

BOILER WATER CHEMISTRY COMPARISON

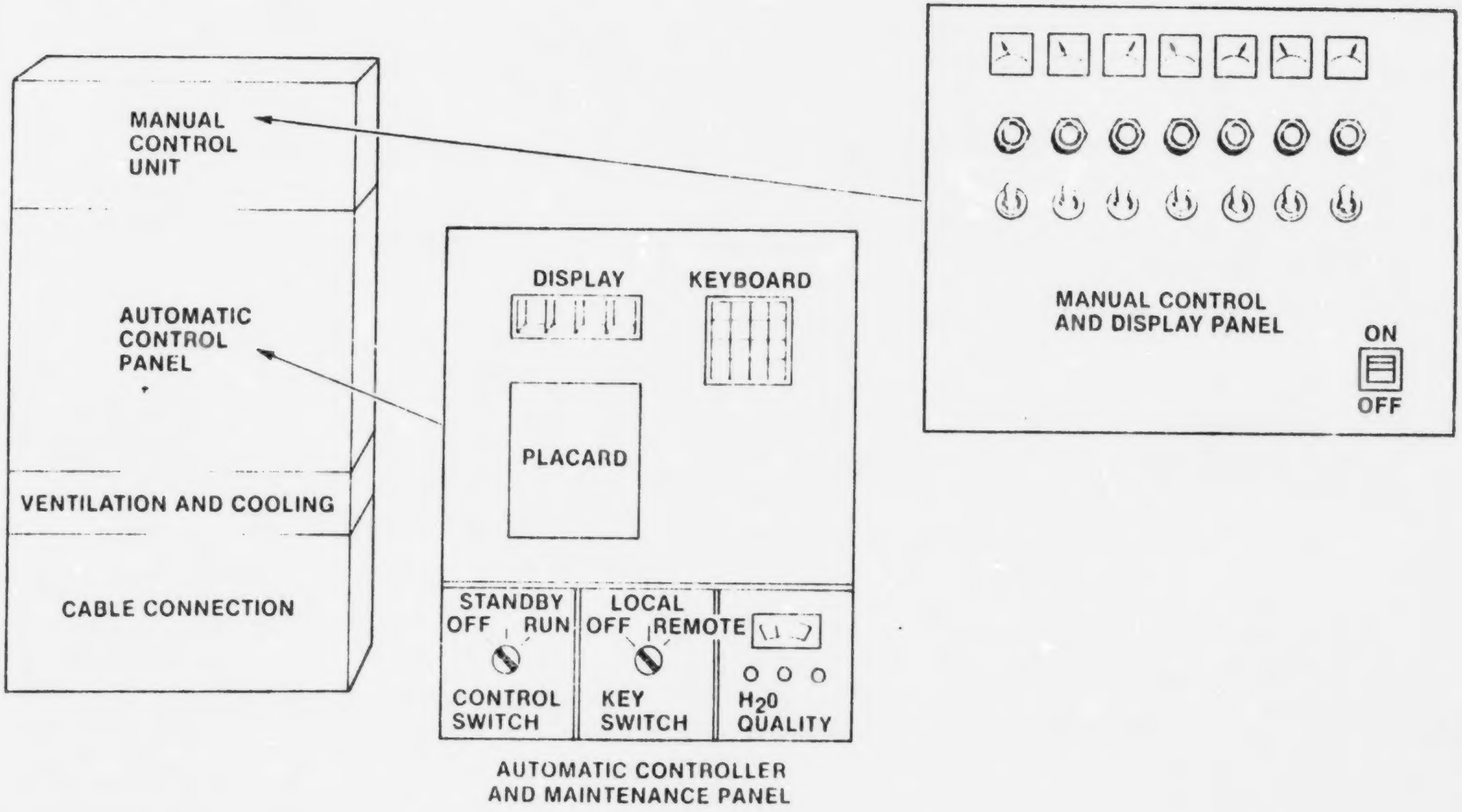
	U.S. NAVY 600 PSI BOILER STANDARD	SOLAR RACER OPERATING PLANT EXPERIENCE*
CONDUCTIVITY (μMHO/CM)	600	0.2
pH	10.2 — 10.6	6 — 7
CHLORIDE (EPM)	1.0	0.00002
SILICA (PPM)	30	0.02
DISSOLVED OXYGEN (PPB)	15	20

* BASED ON OPERATING EXPERIENCE IN SOLAR'S FULL-SCALE DEVELOPMENT FACILITY

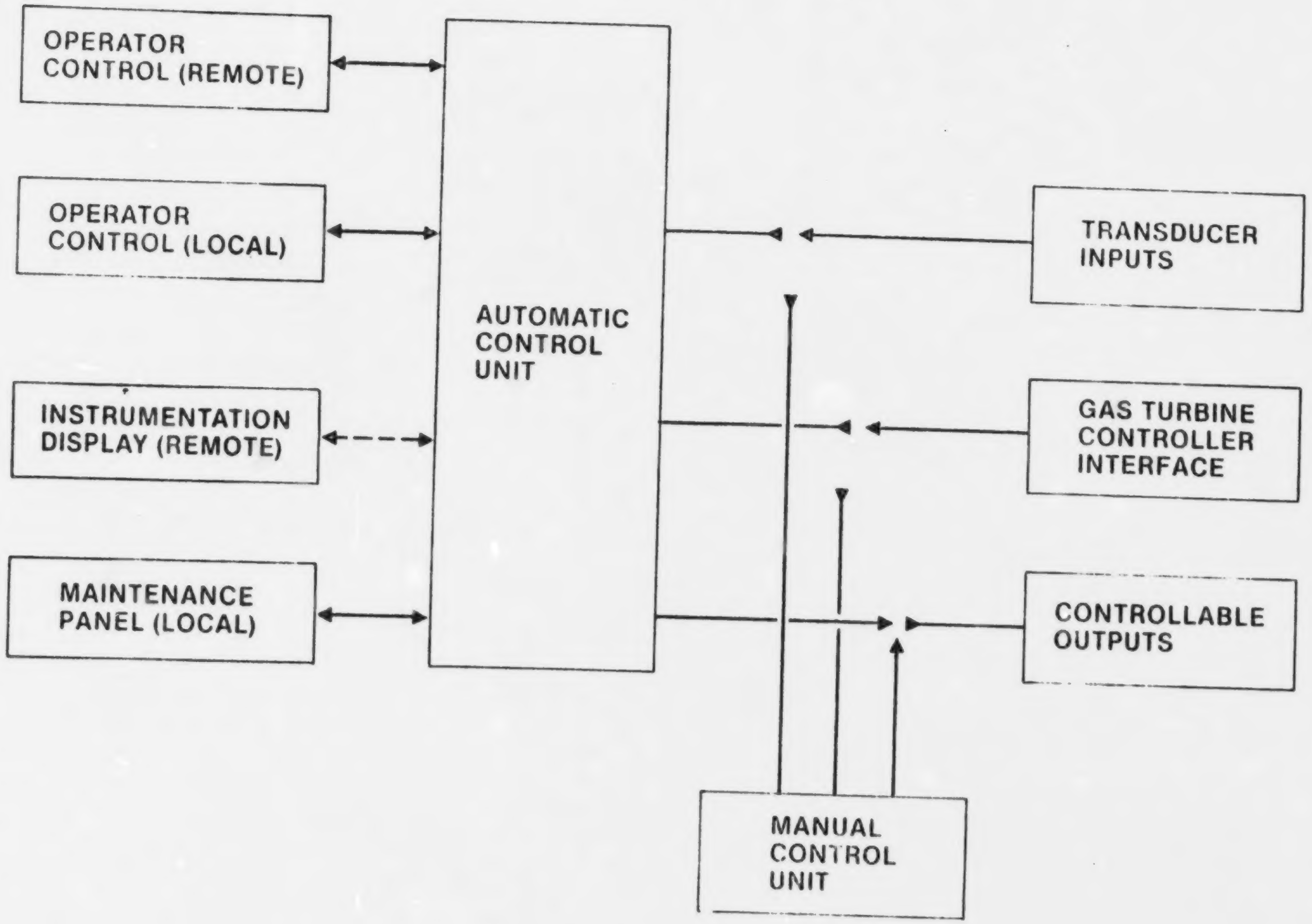
RACER CENTRAL CONTROL OPERATOR PANEL


DISPLAY
CONTROL
REMOTE PANEL
(IN CENTRAL CONTROL)
OPERATOR CONTROL
STANDBY
OFF
RUN
CONTROL
SWITCH
RACER SYSTEM DISPLAY
LOCATOR
1 2 3 4 5 6 7
A
B
C
A B C D
1 4 7 –
2 5 8 –
3 6 9 –
KEYBOARD

RACER ENGINE ROOM (LOCAL) CONTROL PANEL


MANUAL CONTROL UNIT
AUTOMATIC CONTROL PANEL
VENTILATION AND COOLING
CABLE CONNECTION
DISPLAY
KEYBOARD
PLACARD
STANDBY
OFF
RUN
CONTROL SWITCH
LOCAL
OFF
REMOTE
KEY SWITCH
H_20 QUALITY
AUTOMATIC CONTROLLER AND MAINTENANCE PANEL
MANUAL CONTROL AND DISPLAY PANEL
ON
OFF

CONTROL AND INSTRUMENTATION BLOCK DIAGRAM

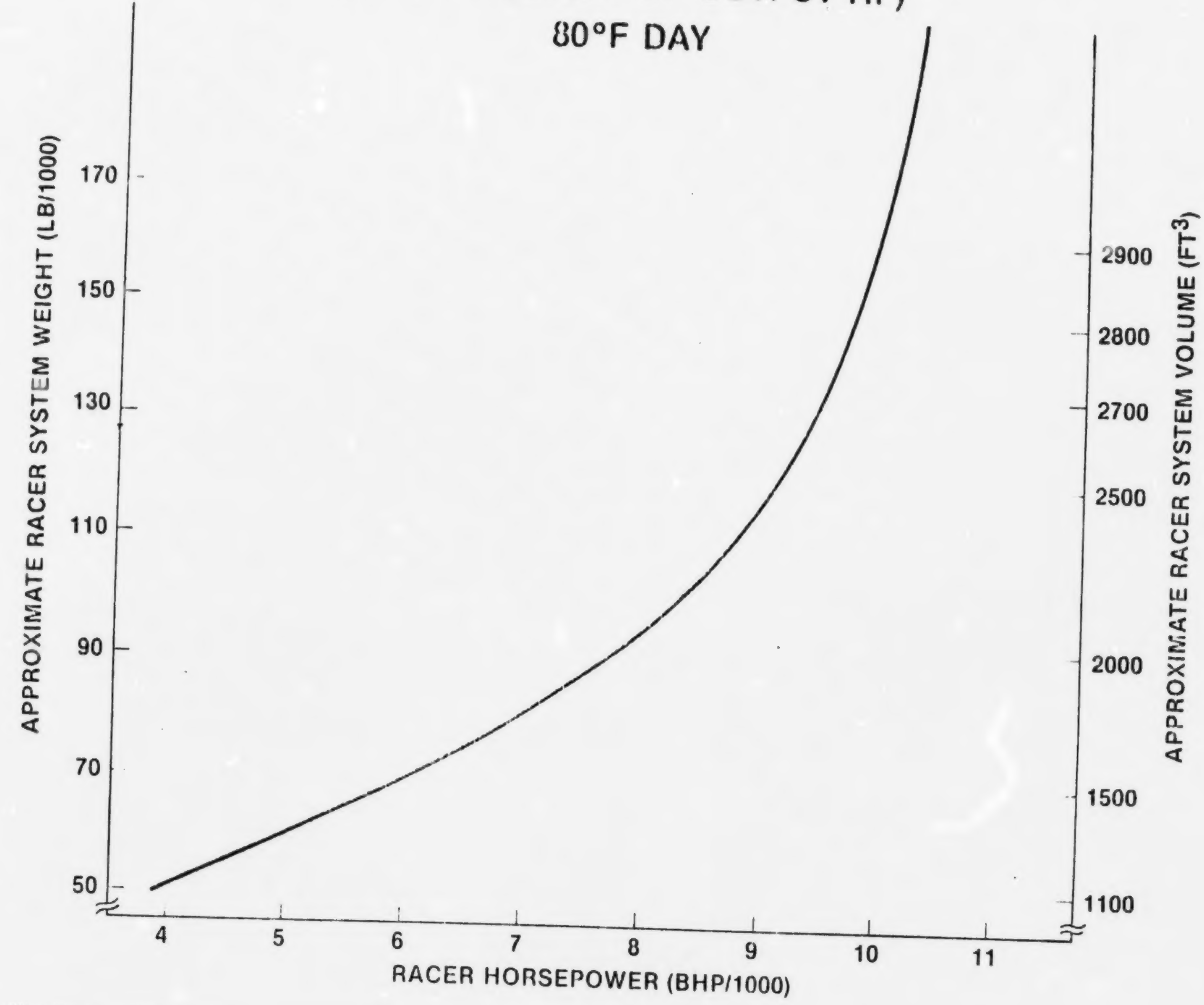

OPERATOR CONTROL (REMOTE)
OPERATOR CONTROL (LOCAL)
INSTRUMENTATION DISPLAY (REMOTE)
MAINTENANCE PANEL (LOCAL)
AUTOMATIC CONTROL UNIT
TRANSDUCER INPUTS
GAS TURBINE CONTROLLER INTERFACE
CONTROLLABLE OUTPUTS
MANUAL CONTROL UNIT

SUMMARY
AUTOMATIC CONTROL UNIT

- TOTAL NUMBER OF IC CHIPS USED — 152
- TOTAL NUMBER OF SEM's USED — 49
- NUMBER OF UNIQUE, APPROVED (FEDERAL SUPPLY SYSTEM) SEM's USED — 16
- NUMBER OF UNIQUE NON-STANDARD ELECTRONIC MODULES USED — 4

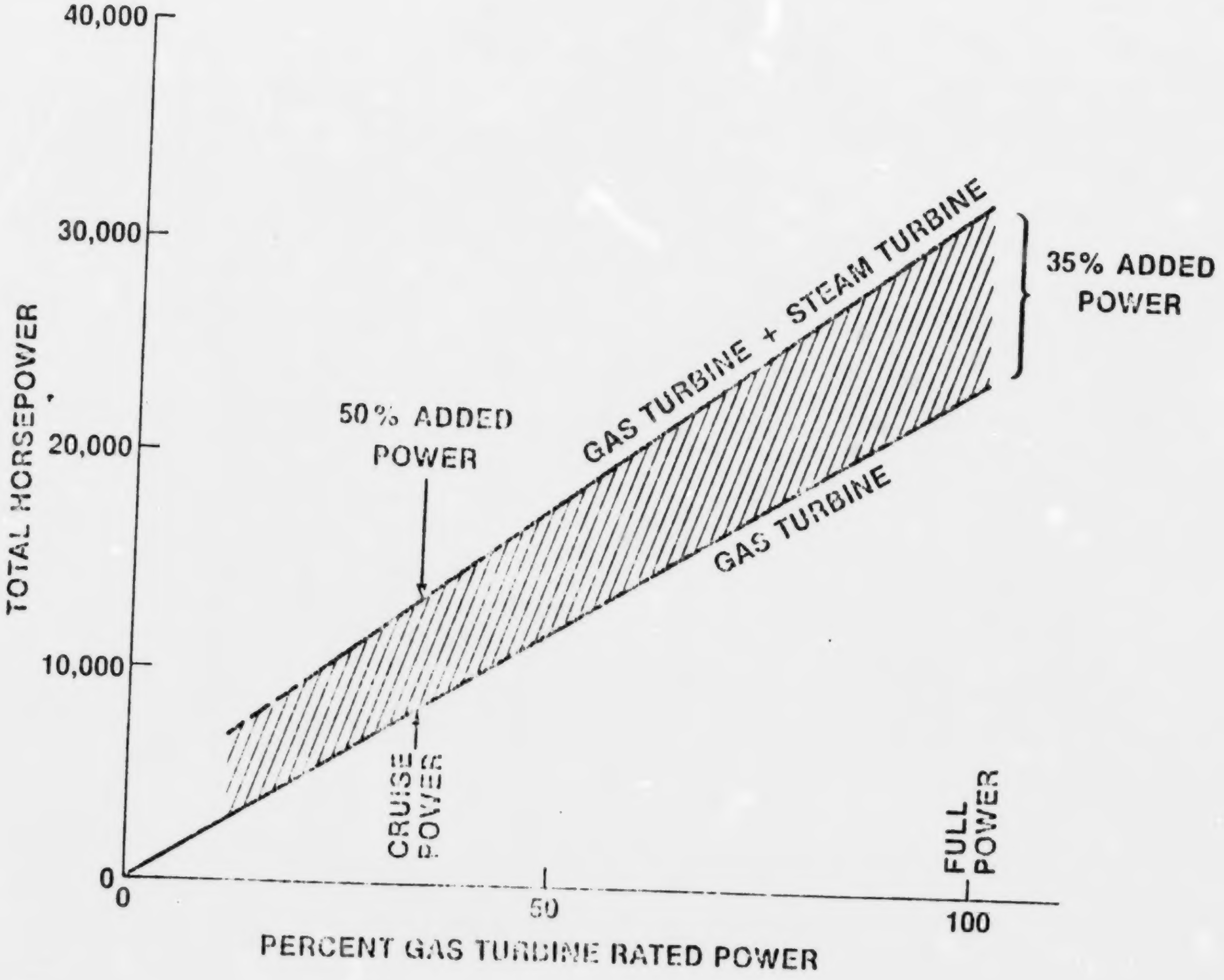
* 80 PERCENT OF AUTOMATIC CONTROL UNIT HARDWARE CONSISTS OF STANDARD ELECTRONIC MODULES (SEM's)

RACER WEIGHT/VOLUME
(AS A FUNCTION OF OUTPUT HP)
80°F DAY



transparency
mounting frame

RACER COMBINED CYCLE POWER SPLIT



Visual Products Division 3M
St. Paul MN 55101 Made in USA

PERFORMANCE SUMMARY
(80°F DAY, $P = f(N^3)$)

	SHIP SPEED (KNOTS)			
	5	20	22	32
RACER OUTPUT HP	306	19,606	26,096	32,306
GT/RACER COMBINED SFC	1.96	0.322	0.311	0.293
FUEL RATIO $\left(\frac{\text{LM2500 ONLY}}{\text{LM2500 WITH RACER}}\right)$	2.09	1.55	1.51	1.17

Visual Products Division 3M

RACER SYSTEM ESTIMATED WEIGHTS AND VOLUMES

COMPONENT	DRY WEIGHT (LB)	VOLUME (FT^3)
• WASTE HEAT BOILER		
— CORE	22,240	478
— STRUCTURE	13,200	493
• STACK HEATER		
— CORE	4,200	68
— STRUCTURE	2.380	160
• STEAM TURBINE	16,400	362
• CONDENSER (DRY)	15,431	420
• CONTROLS & INSTRUMENTATION	400	40
• REDUCTION GEAR & CLUTCH	4,407	47
• AUXILIARY SYSTEMS		
— WATER TREATMENT SUBSYSTEM	1,100	16
— STEAM SUBSYSTEM	2,705	45
— CONDENSATE SUBSYSTEM	2,791	69.4
— FEEDWATER SUBSYSTEM	7,737	84.0
— CONDENSER VACUUM SUBSYSTEM	1,909	25.0
— CHLORINATOR SUBSYSTEM	725	20.0
— SEALS AND BEARING SUPPLY	136	6
• MISCELLANEOUS	4669	256
TOTAL	98,050	2566

ransparency
nounting frame

RACER SYSTEM PERFORMANCE
AT FULL POWER

AMBIENT TEMP.	GT HP	STEAM HP	TOTAL HP	HP x 2	SFC (LB/HP-HR)
100°F	23,590	9,310	32,900	65,800	.28
80°F	27,150	9,510	36,660	73,320	.28
59°F	29,200	9,670	38,870	77,740	.28

RANKINE CYCLE ENERGY RECOVERY
(RACER)

U.S. NAVY PROGRAM
APPLIED TO LM2500 ENGINE

31-3809

OCTOBER 9, 1980

RACER OBJECTIVES APPLICABLE TO MELVILLE SHIPPING, LTD.

- MINIMUM WEIGHT AND VOLUME
- HIGH RELIABILITY WITH MINIMUM MAINTENANCE

  MTBF $\geqslant$ 10,000 HOURS
  LESS THAN 50 MAINTENANCE MANHOURS PER WEEK
- AUTOMATIC CONTROL SYSTEM FOR UNMANNED OPERATION
- CAPABLE OF DRY FIRING

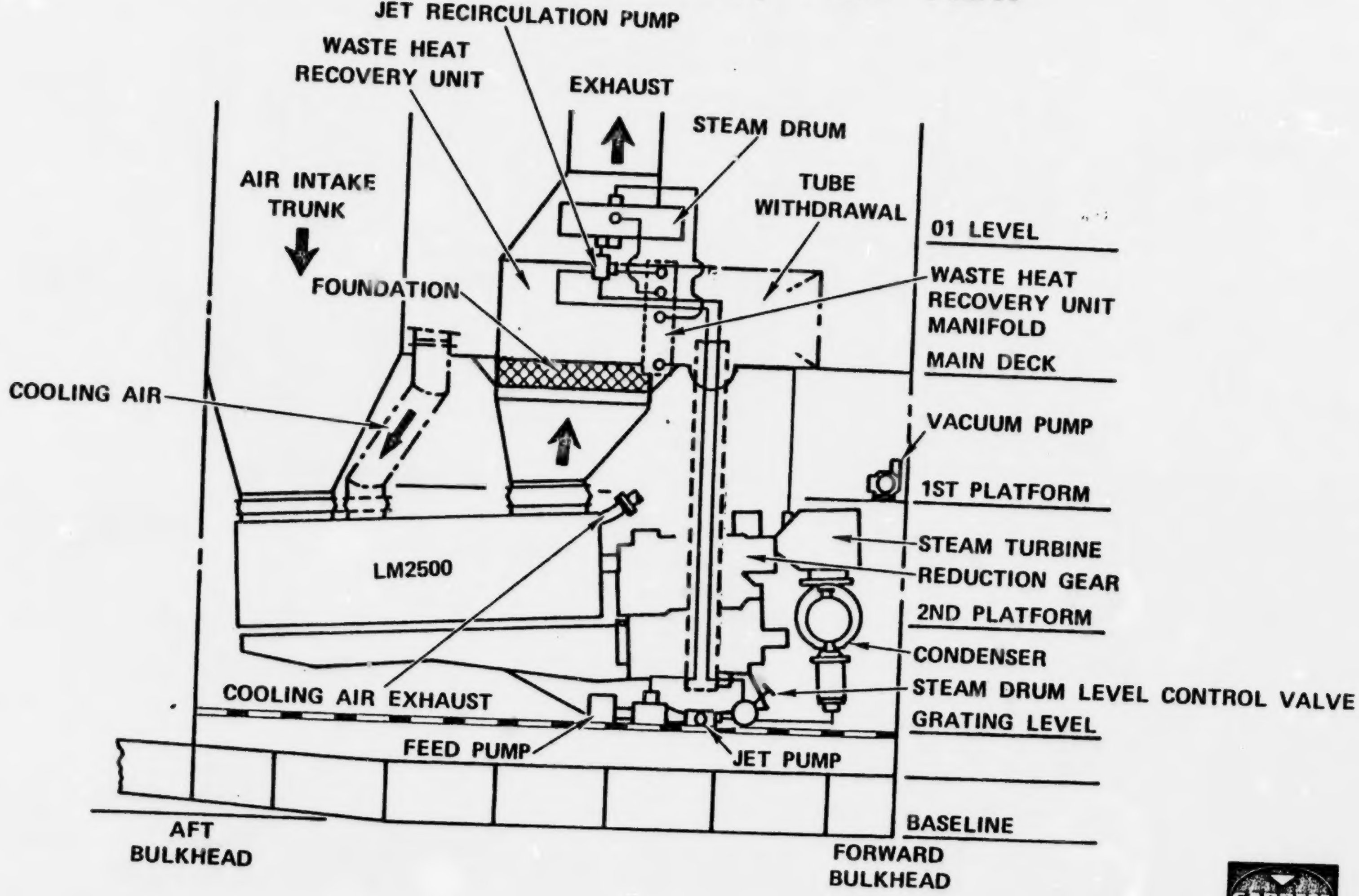
AIRESEARCH RACER APPROACH

- ACHIEVE HIGH RELIABILITY THROUGH SIMPLICITY
  - REVIEW CONVENTIONAL STEAM SYSTEMS AND ELIMINATE PROBLEMS FIRST – SOLVE PROBLEMS SECOND
- USE ADVANCED TECHNOLOGY AND SYSTEMS DESIGN METHODS

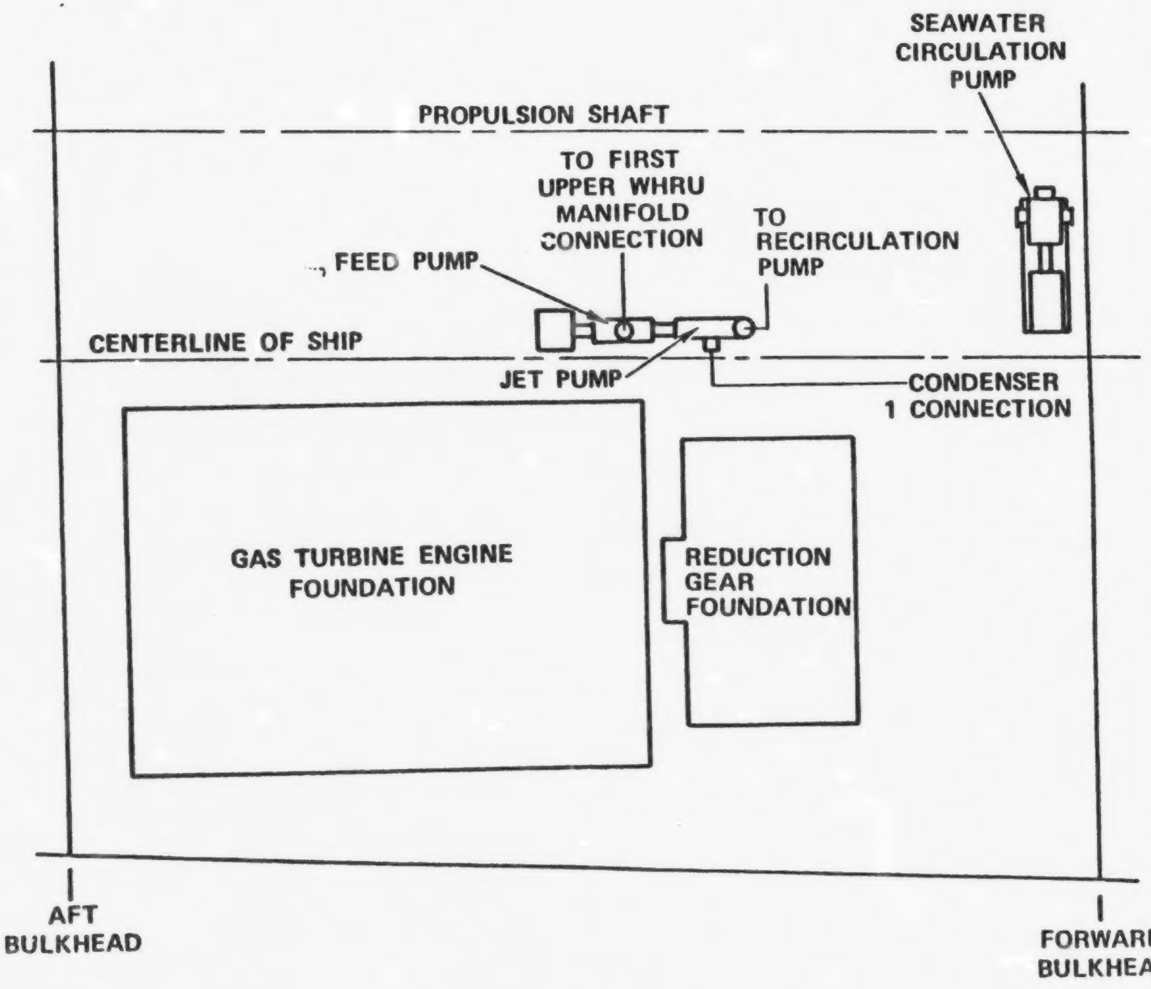
OBJECTIVES ACHIEVED BY:

- HIGH TECHNOLOGY WHRU AND STEAM TURBINE
  - ○ TOTAL WEIGHT WITH FOUNDATIONS = 120,000 POUNDS
  - ○ TURBINE EFFICIENCY = 0.87
- HIGH RELIABILITY THROUGH ELIMINATING PROBLEMS ASSOCIATED WITH CONVENTIONAL STEAM SYSTEMS
  - ○ SINGLE, CONSTANT SPEED, FEEDWATER PUMP SYSTEM THAT ELIMINATES SURGE AND RETURN-WATER PROBLEMS
  - ○ CORROSION RESISTANT MATERIALS USED THROUGHOUT, ELIMINATES NEED FOR CHEMICAL WATER TREATMENT
  - ○ AN ESSENTIALLY HERMETICALLY SEALED STEAM SYSTEM TO MINIMIZE NEED FOR MAKEUP WATER (2 GALLONS PER DAY)
  - ○ WATER LUBRICATED BEARINGS TO ELIMINATE OIL CONTAMINATION AND SEALING PROBLEMS
  - ○ MTBF = 10,223 HOURS
  - ○ SHIPBOARD MAINTENANCE = 5.9 MANHOURS PER WEEK
- FULLY AUTOMATIC, SELF-REGULATING, CONTROL SYSTEM MANUAL CAPABILITY AVAILABLE
- WHRU CONSTRUCTED OF IN800 MATERIAL, ALLOWS DRY FIRING
  PEAK GAS TURBINE EXHAUST TEMPERATURE = 1060°F

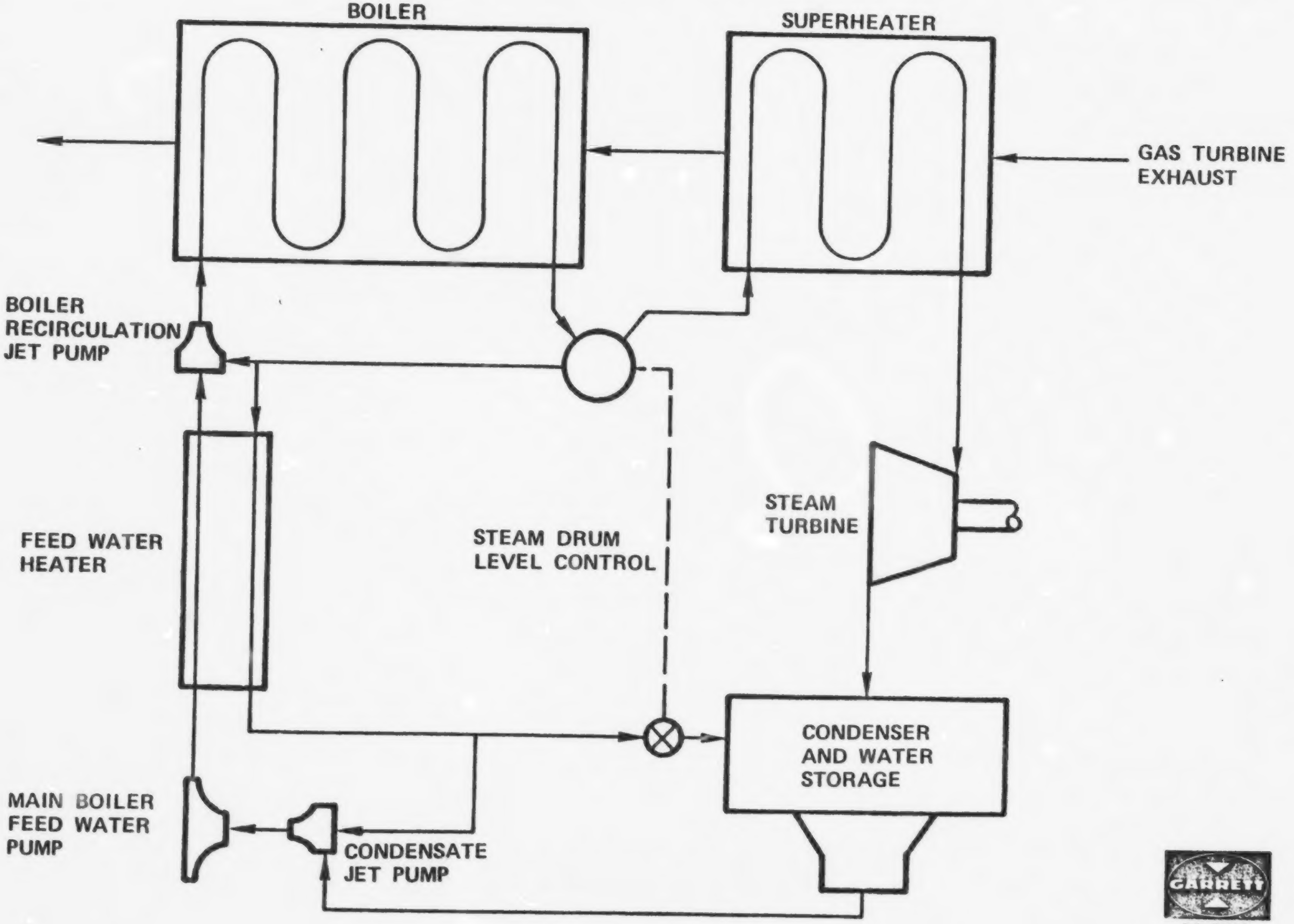
RACER INSTALLATION IN DESTROYER SHOWING PIPING, SIDE VIEW


JET RECIRCULATION PUMP
WASTE HEAT RECOVERY UNIT
EXHAUST
STEAM DRUM
TUBE WITHDRAWAL
AIR INTAKE TRUNK
01 LEVEL
FOUNDATION
WASTE HEAT RECOVERY UNIT MANIFOLD
MAIN DECK
COOLING AIR
VACUUM PUMP
1ST PLATFORM
STEAM TURBINE
LM2500
REDUCTION GEAR
2ND PLATFORM
CONDENSER
STEAM DRUM LEVEL CONTROL VALVE
COOLING AIR EXHAUST
GRATING LEVEL
FEED PUMP
JET PUMP
BASELINE
AFT BULKHEAD
FORWARD BULKHEAD

RACER INSTALLATION IN DESTROYER, TOP VIEW


SEAWATER CIRCULATION PUMP
PROPULSION SHAFT
TO FIRST UPPER WHRU MANIFOLD CONNECTION
TO RECIRCULATION PUMP
FEED PUMP
CENTERLINE OF SHIP
JET PUMP
CONDENSER 1 CONNECTION
GAS TURBINE ENGINE FOUNDATION
REDUCTION GEAR FOUNDATION
AFT BULKHEAD
FORWARD BULKHEAD

RACER FLOW SCHEMATIC


BOILER
SUPERHEATER
GAS TURBINE EXHAUST
BOILER RECIRCULATION JET PUMP
STEAM TURBINE
FEED WATER HEATER
STEAM DRUM LEVEL CONTROL
CONDENSER AND WATER STORAGE
MAIN BOILER FEED WATER PUMP
CONDENSATE JET PUMP

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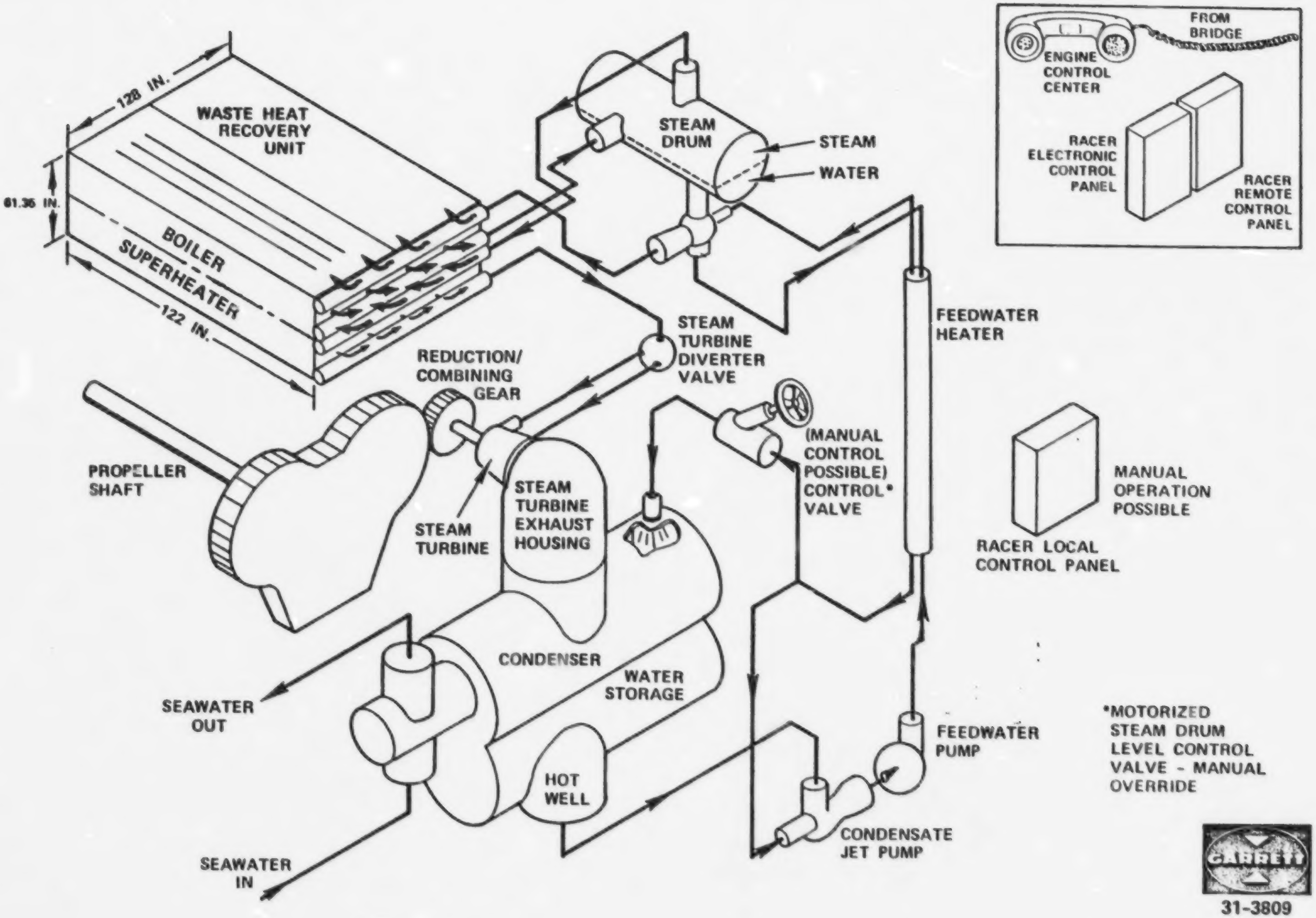
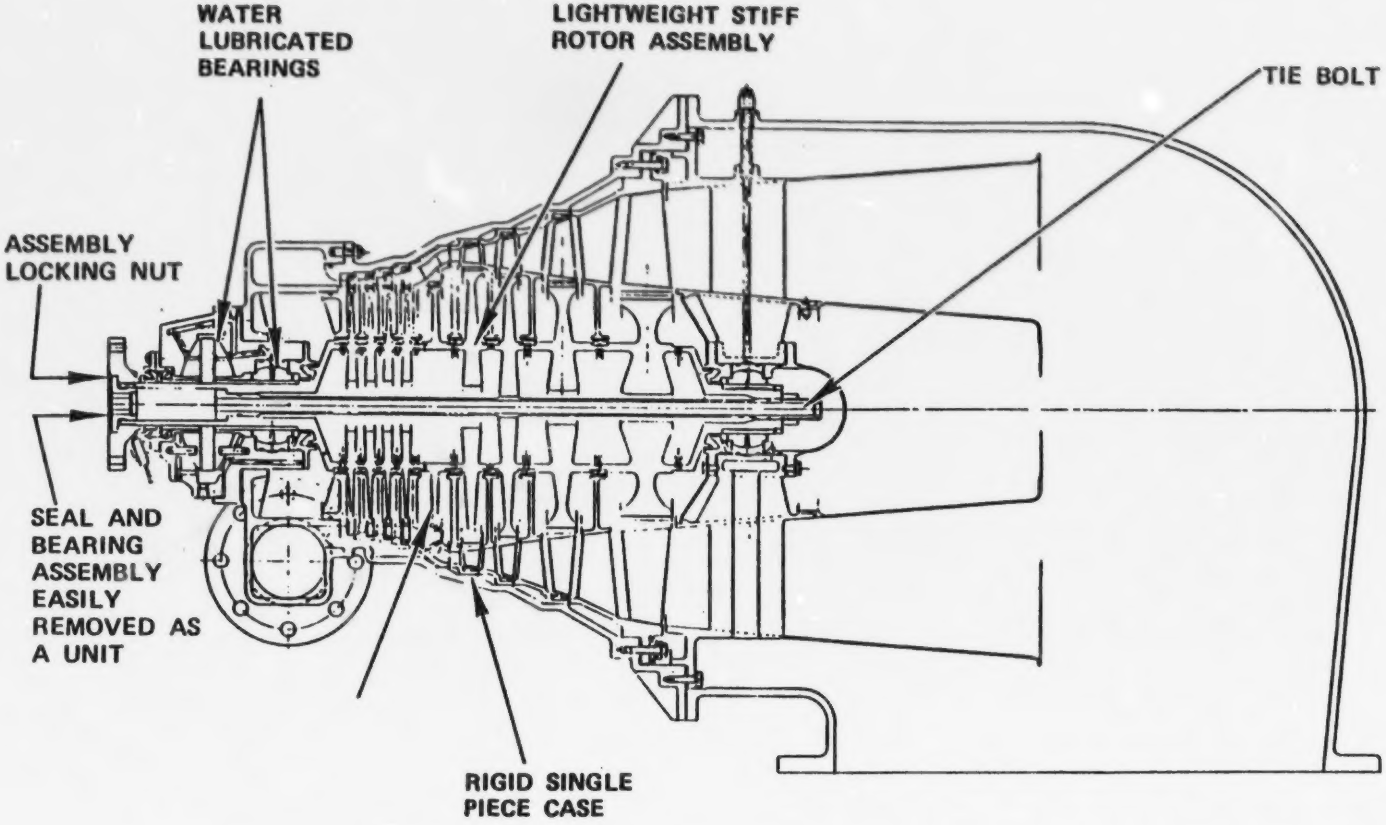
RACER SYSTEM SCHEMATIC


FROM BRIDGE
ENGINE CONTROL CENTER
RACER ELECTRONIC CONTROL PANEL
RACER REMOTE CONTROL PANEL
128 IN.
WASTE HEAT RECOVERY UNIT
61.35 IN.
BOILER
SUPERHEATER
122 IN.
STEAM DRUM
STEAM
WATER
FEEDWATER HEATER
STEAM TURBINE DIVERTER VALVE
REDUCTION/ COMBINING GEAR
PROPELLER SHAFT
STEAM TURBINE
STEAM TURBINE EXHAUST HOUSING
(MANUAL CONTROL POSSIBLE) CONTROL* VALVE
RACER LOCAL CONTROL PANEL
MANUAL OPERATION POSSIBLE
CONDENSER
WATER STORAGE
SEAWATER OUT
HOT WELL
SEAWATER IN
FEEDWATER PUMP
CONDENSATE JET PUMP

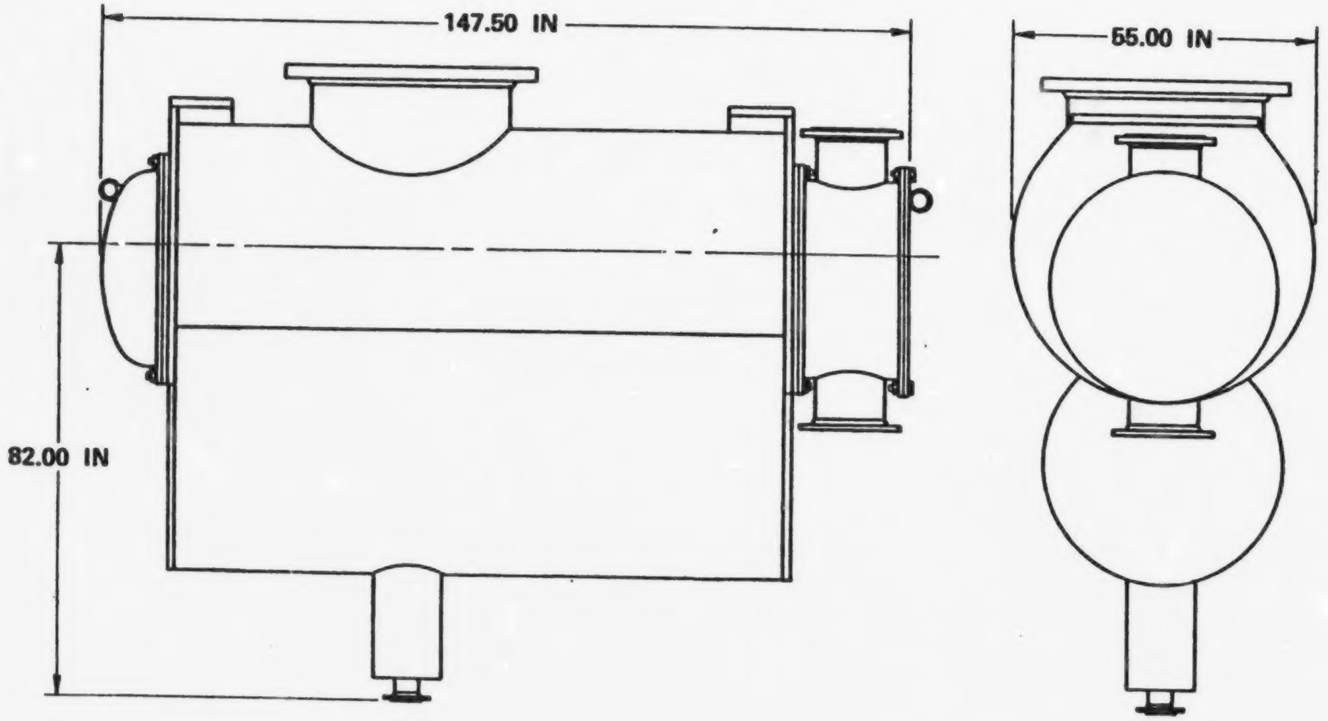
*MOTORIZED STEAM DRUM LEVEL CONTROL VALVE - MANUAL OVERRIDE

31-3809
7

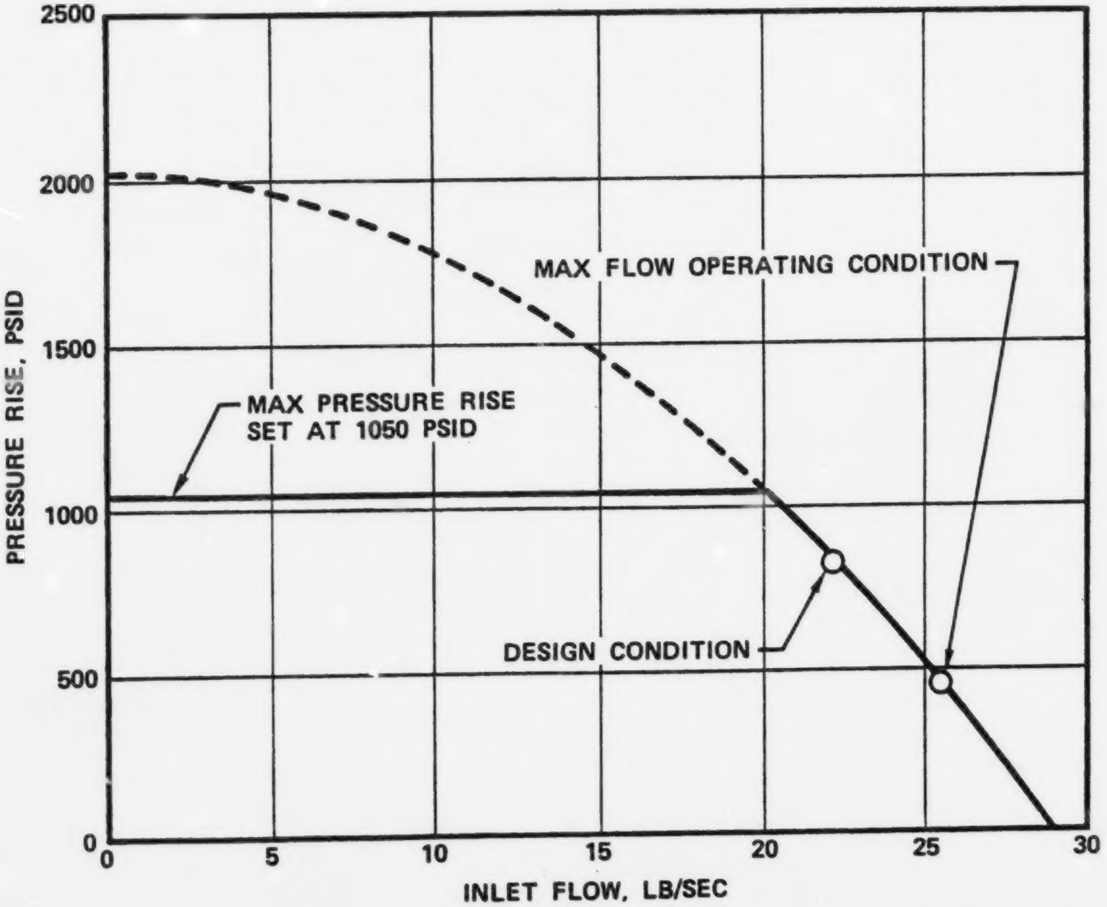
RACER STEAM TURBINE


WATER LUBRICATED BEARINGS
LIGHTWEIGHT STIFF ROTOR ASSEMBLY
TIE BOLT
ASSEMBLY LOCKING NUT
SEAL AND BEARING ASSEMBLY EASILY REMOVED AS A UNIT
RIGID SINGLE PIECE CASE

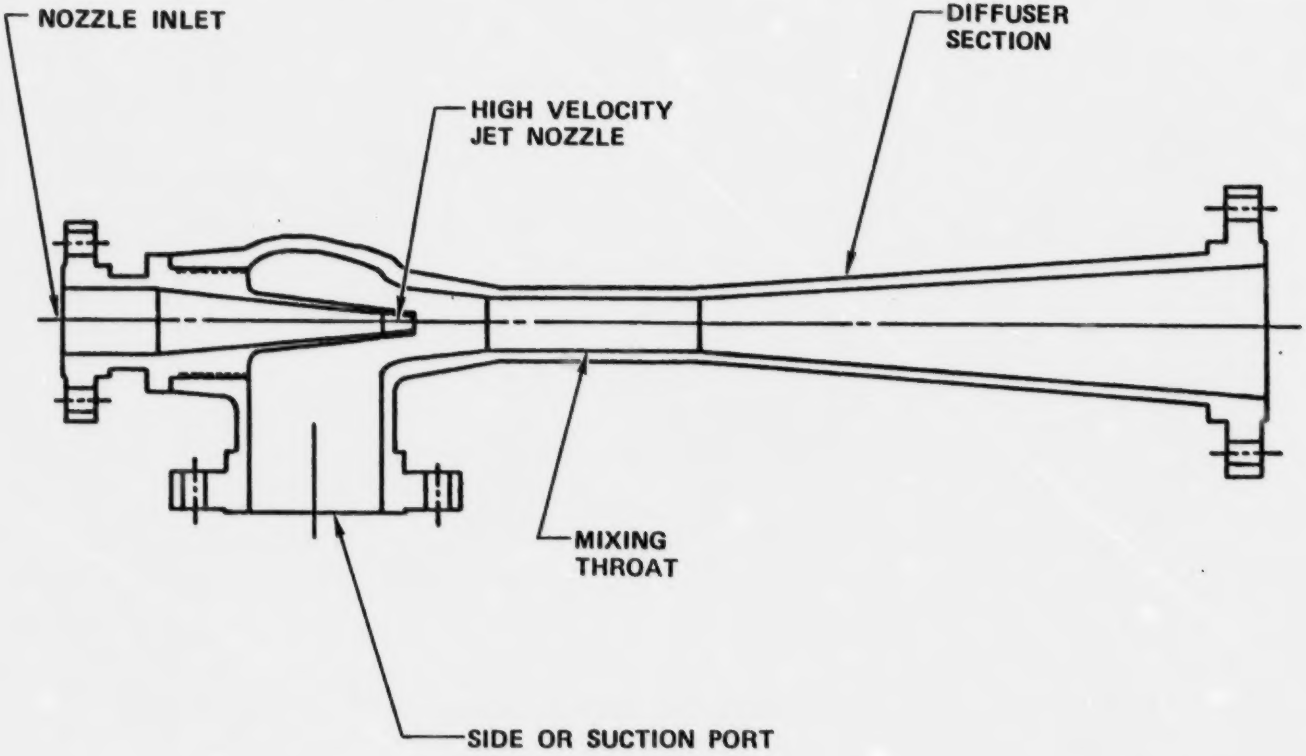
CONDENSER


147.50 IN
55.00 IN
82.00 IN

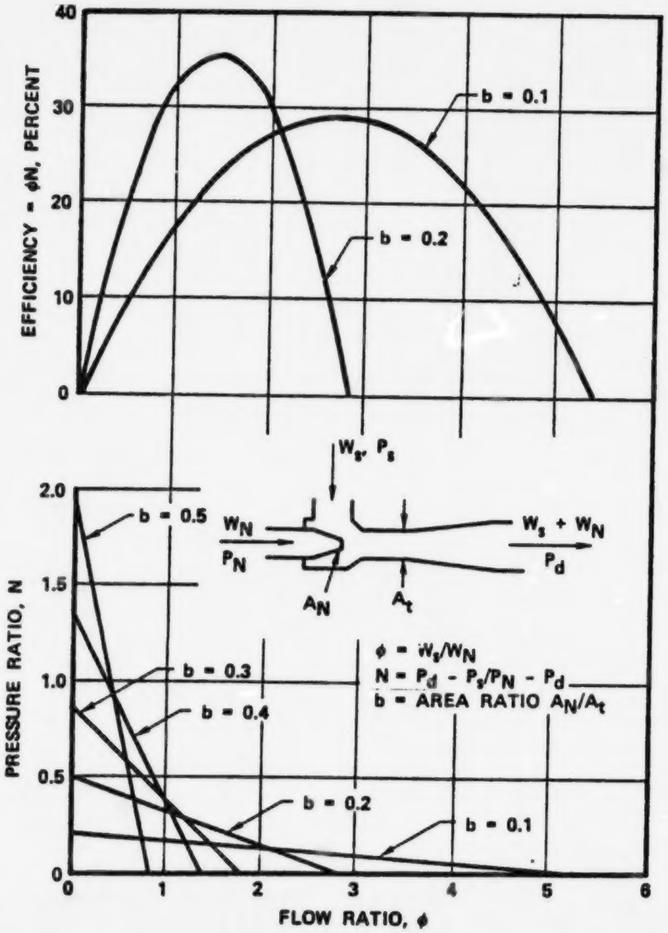
BOILER FEEDPUMP CHARACTERISTICS



TYPICAL WATER JET PUMP

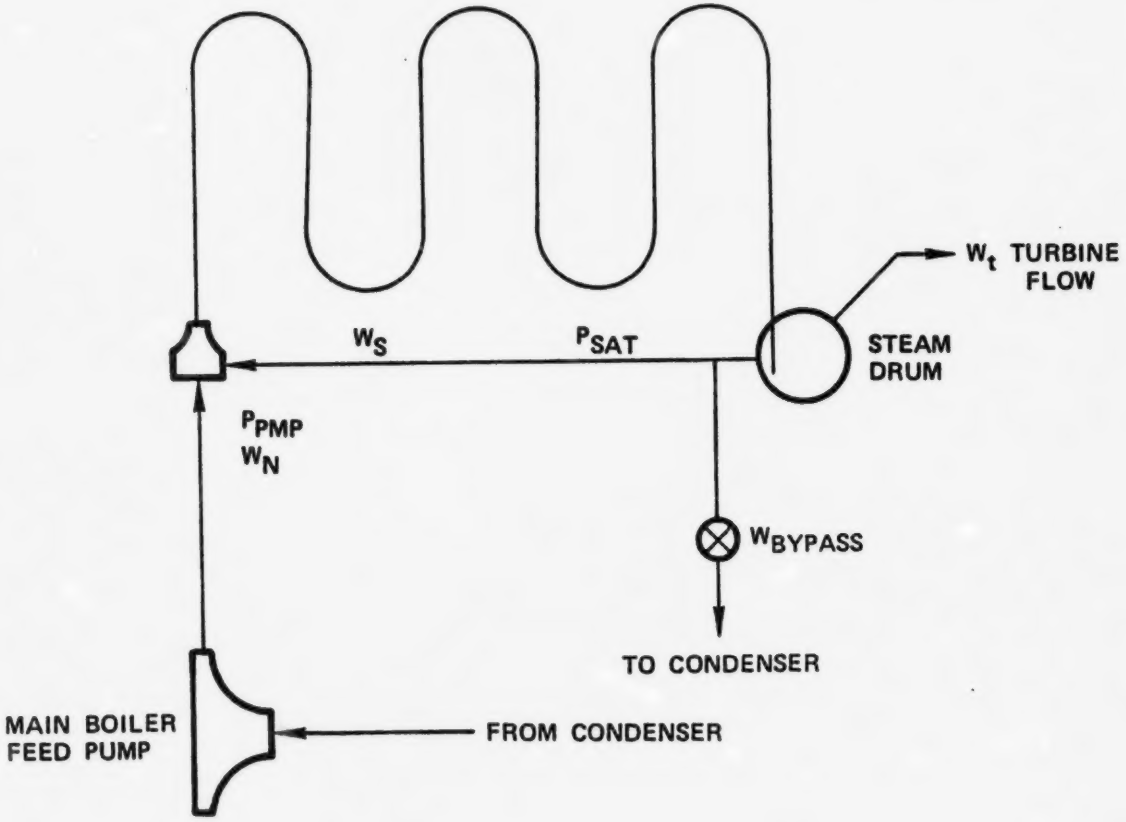

NOZZLE INLET
HIGH VELOCITY JET NOZZLE
DIFFUSER SECTION
MIXING THROAT
SIDE OR SUCTION PORT

WATER JET PUMP CHARACTERISTICS

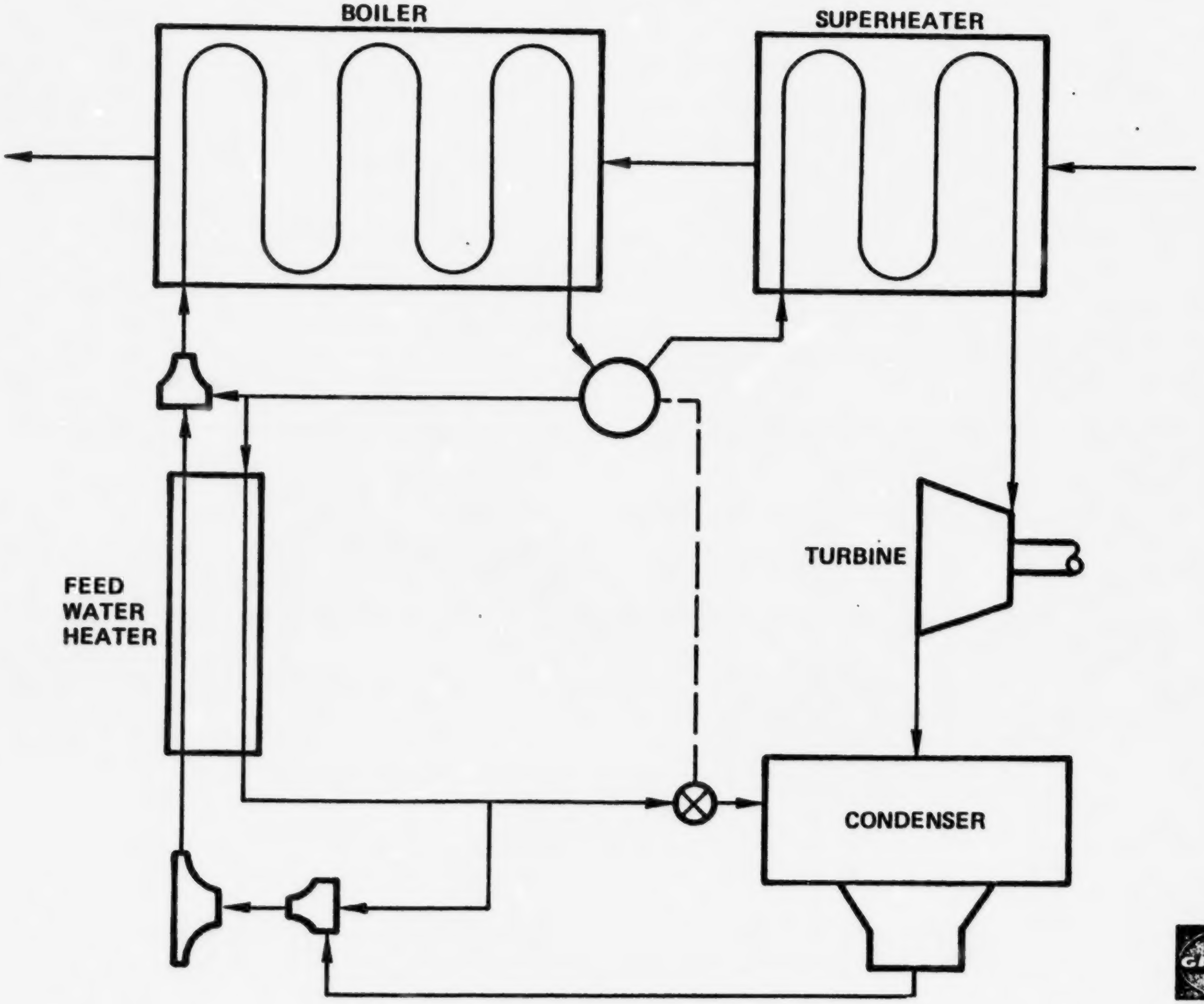


$\phi = W_s/W_N$
$N = P_d - P_s/P_N - P_d$
b = AREA RATIO A_N/A_t

MINIMUM RACER PUMP AND CONTROL SYSTEM


W_t TURBINE FLOW
STEAM DRUM
W_S
P_{SAT}
P_{PMP}
W_N
W_{BYPASS}
TO CONDENSER
MAIN BOILER FEED PUMP
FROM CONDENSER

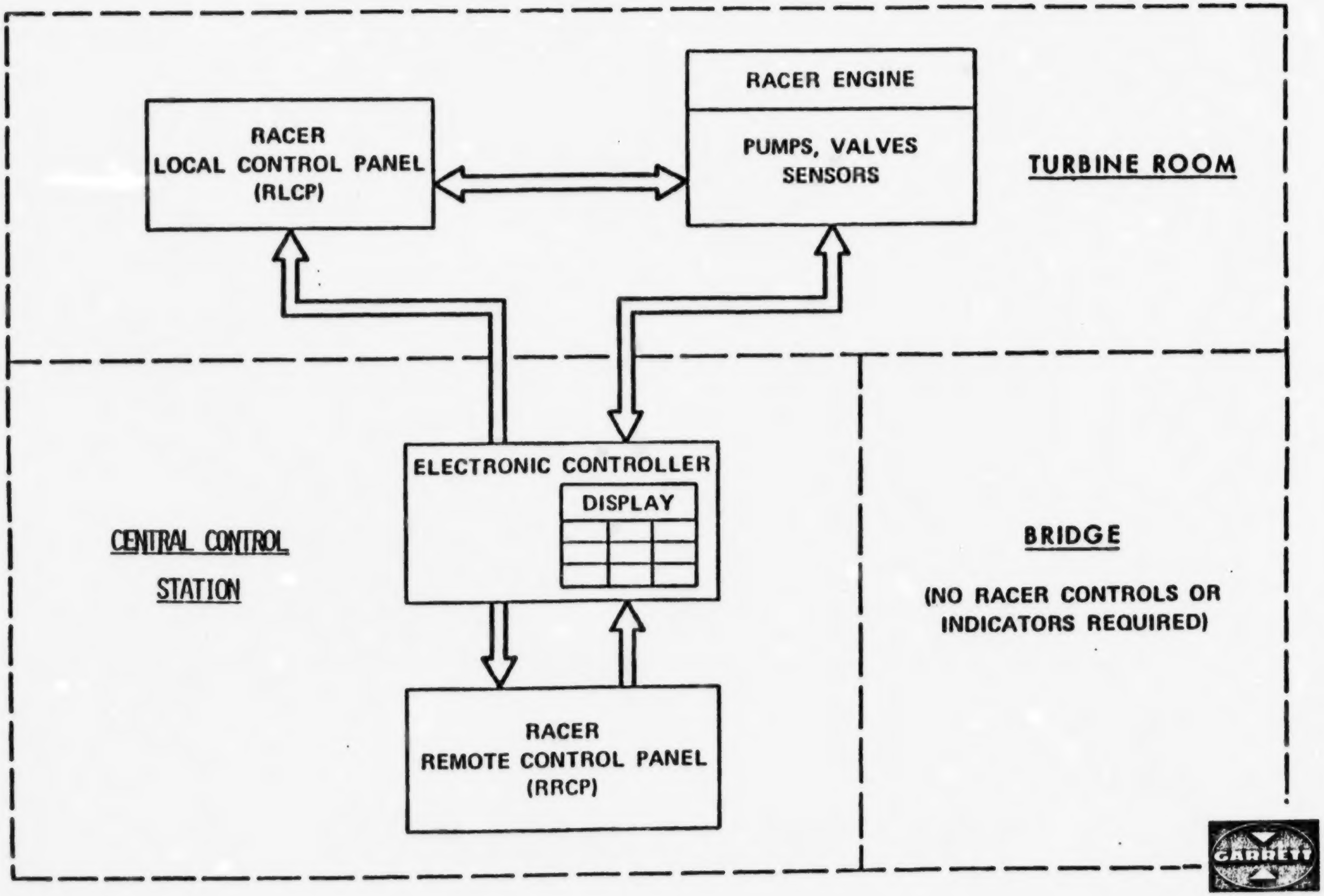
RACER SYSTEM SCHEMATIC


BOILER
SUPERHEATER
TURBINE
FEED
WATER
HEATER
CONDENSER

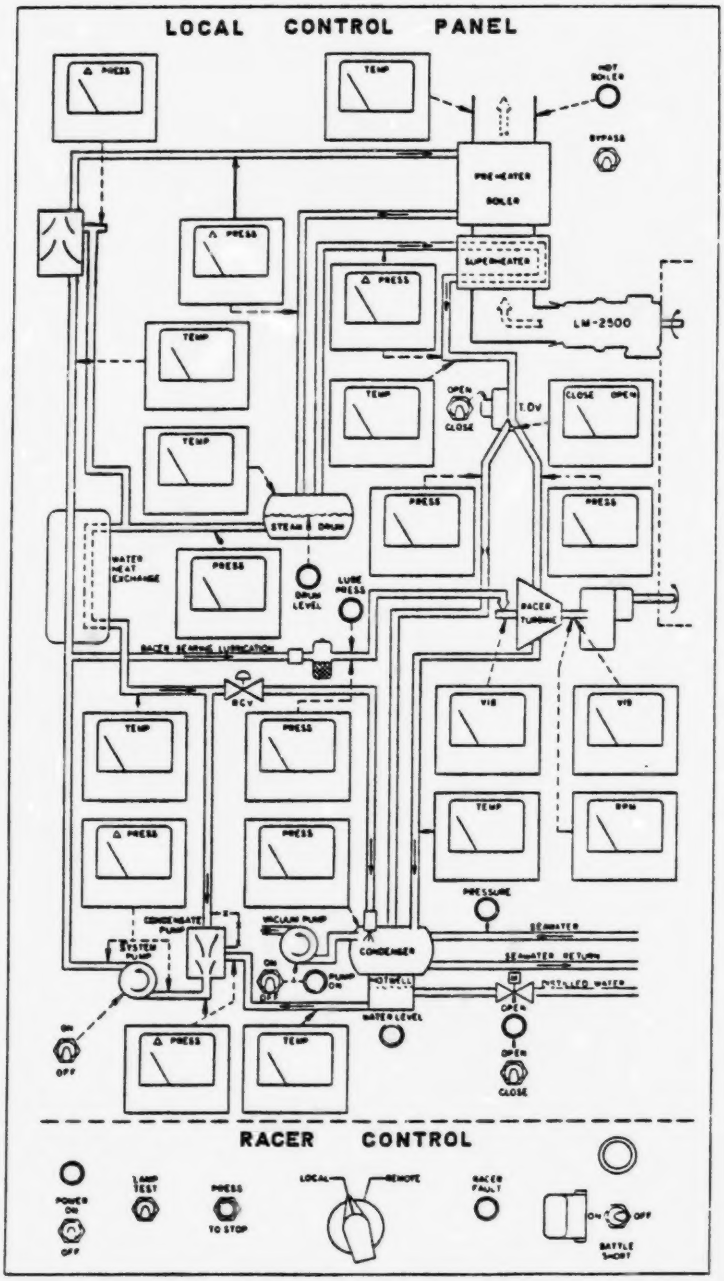
RACER CONTROL SYSTEM SUMMARY

- PROVIDES AUTOMATIC STARTUP OR SHUTDOWN OF RACER
- PROVIDES HEALTH MONITORING OF RACER SYSTEM
- INCLUDES BUILT-IN-TEST-EQUIPMENT TO AID IN CONTROL SYSTEM TROUBLESHOOTING
- NORMAL OPERATION OF RACER INDEPENDENT OF CONTROL AND MONITORING SYSTEMS
- MANUAL OPERATION OF RACER AVAILABLE THROUGH BYPASSING OF AUTOMATIC CONTROL FUNCTIONS
- CONSISTS OF ENGINE ROOM (LOCAL) AND CENTRAL CONTROL ROOM (REMOTE) CONTROL PANELS

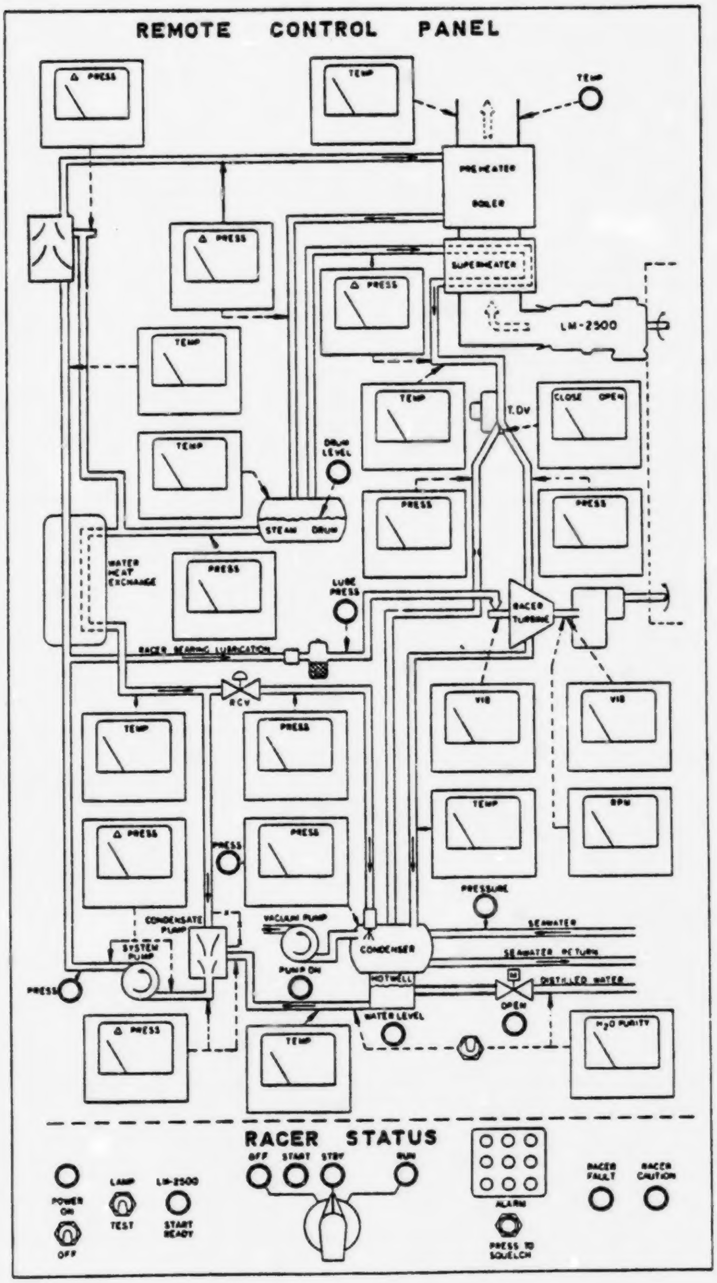
CONTROL COMPONENTS LOCATION BLOCK DIAGRAM


RACER
LOCAL CONTROL PANEL
(RLCP)
RACER ENGINE
PUMPS, VALVES
SENSORS
TURBINE ROOM
ELECTRONIC CONTROLLER
DISPLAY
CENTRAL CONTROL
STATION
BRIDGE
(NO RACER CONTROLS OR
INDICATORS REQUIRED)
RACER
REMOTE CONTROL PANEL
(RRCP)

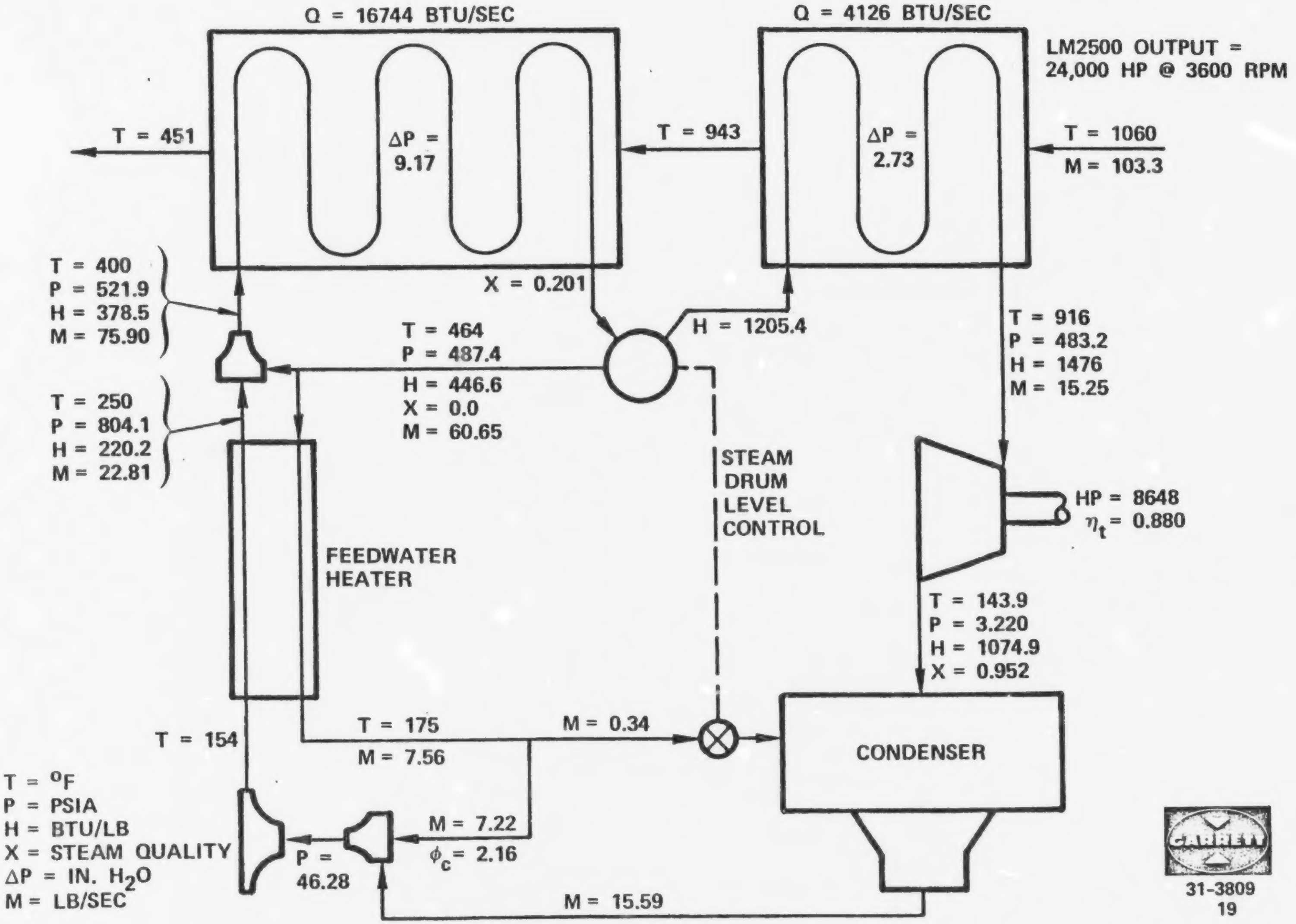
RACER LOCAL CONTROL PANEL


LOCAL CONTROL PANEL
Δ PRESS
TEMP
HOT BOILER
BYPASS
PREHEATER BOILER
Δ PRESS
SUPERHEATER
Δ PRESS
LM-2500
TEMP
TEMP
TEMP
OPEN
CLOSE
T.DV
CLOSE OPEN
STEAM DRUM
PRESS
PRESS
WATER HEAT EXCHANGE
PRESS
DRUM LEVEL
LUBE PRESS
RACER TURBINE
RACER BEARING LUBRICATION
RCV
TEMP
PRESS
VIB
VIB
Δ PRESS
PRESS
TEMP
RPM
PRESSURE
CONDENSATE PUMP
VACUUM PUMP
SYSTEM PUMP
CONDENSER
SEAWATER
SEAWATER RETURN
DISTILLED WATER
ON
OFF
PUMP ON
HOTWELL
WATER LEVEL
OPEN
OPEN
CLOSE
ON
OFF
Δ PRESS
TEMP
RACER CONTROL
POWER ON
OFF
LAMP TEST
PRESS
TO STOP
LOCAL
REMOTE
RACER FAULT
ON
OFF
BATTLE SHORT

RACER REMOTE CONTROL PANEL


REMOTE CONTROL PANEL
Δ PRESS
TEMP
TEMP
PREHEATER
BOILER
SUPERHEATER
LM-2500
Δ PRESS
Δ PRESS
TEMP
TEMP
TEMP
T.DV
CLOSE OPEN
DRUM LEVEL
STEAM DRUM
PRESS
PRESS
WATER HEAT EXCHANGE
PRESS
LUBE PRESS
RACER TURBINE
RACER BEARING LUBRICATION
RCV
TEMP
PRESS
VIB
VIB
Δ PRESS
PRESS
PRESS
TEMP
RPM
PRESSURE
CONDENSATE PUMP
VACUUM PUMP
CONDENSER
SEAWATER
SEAWATER RETURN
SYSTEM PUMP
PRESS
PUMP ON
HOTWELL
DISTILLED WATER
OPEN
WATER LEVEL
Δ PRESS
TEMP
H_2O PURITY
RACER STATUS
OFF
START
STBY
RUN
POWER ON
OFF
LAMP TEST
LM-2500 START READY
ALARM
PRESS TO SQUELCH
RACER FAULT
RACER CAUTION

RACER SYSTEM STATE POINTS


Q = 16744 BTU/SEC
Q = 4126 BTU/SEC
LM2500 OUTPUT = 24,000 HP @ 3600 RPM
T = 451
ΔP = 9.17
T = 943
ΔP = 2.73
T = 1060
M = 103.3
X = 0.201
H = 1205.4
T = 400
P = 521.9
H = 378.5
M = 75.90
T = 464
P = 487.4
H = 446.6
X = 0.0
M = 60.65
T = 916
P = 483.2
H = 1476
M = 15.25
T = 250
P = 804.1
H = 220.2
M = 22.81
STEAM DRUM LEVEL CONTROL
HP = 8648
η_t = 0.880
FEEDWATER HEATER
T = 143.9
P = 3.220
H = 1074.9
X = 0.952
T = 175
M = 7.56
M = 0.34
T = 154
CONDENSER
M = 7.22
φ_c = 2.16
P = 46.28
M = 15.59
T = °F
P = PSIA
H = BTU/LB
X = STEAM QUALITY
ΔP = IN. H2O
M = LB/SEC
GARRETT

31-3809

RACER PERFORMANCE VARIATIONS

	FOR U.S. NAVY	FOR MELVILLE SHIPPING	
		AS IS	SUPPLEMENT FIRED
AMBIENT TEMPERATURE, °F	100	20	20
SEAWATER COOLANT TEMP, °F	85	35	35
LM2500 CONDITIONS			
EXHAUST TEMP, °F	1,060	894	1,060
AIRFLOW, LB/SEC	103.3	153.95	153.95
OUTPUT POWER, HP	24,000	28,000	28,000
STEAM TURBINE CONDITIONS			
INLET TEMPERATURE, °F	916	807	904
INLET PRESSURE, PSIA	483	407	562
STEAM FLOW, LB/SEC	15.25	13.40	17.82
OUTPUT POWER, HP	8,648	7,767	10,701
TOTAL OUTPUT POWER, HP	32,648	35,767	38,701
FUEL CONSUMPTION, LB/HR-HP	0.3004	0.3000	0.3128

MELVILLE SHIPPING QUESTIONS

A. WASTE HEAT BOILER DESIGN

1. BOILER STATE POINTS? SEE NAVY RACER SCHEMATIC

2. NUMBER OF BOILERS?
   - ONE RACER SYSTEM PER LM2500
   - TOTAL OUTPUT POWER = 214,602 HORSEPOWER
   - RACER DESIGNED FOR AUTOMATIC OPERATION ON ONE-FOR-ONE BASIS

3. SUPPLEMENTARY FIRING?
   - FEASIBILITY DEMONSTRATED (NOT BY AIRESEARCH)
   - ADDITIONAL OUTPUT = 2934 HORSEPOWER
   - INCREASED SFC, FROM 0.3000 TO 0.3128 LB/HP-HR BASED ON NO INCREASED SURFACE AREA
   - LM2500 BACKPRESSURE INCREASED BY 1.2 IN H_2O

4. DRY FIRING?
   - RACER DESIGNED FOR DRY FIRING
   - ELIMINATES NEED FOR DAMPER AND EXHAUST GAS BYPASS
   - WILL REDUCE LM2500 OUTPUT LIMIT DUE TO INCREASED BACKPRESSURE

B. WASTE HEAT BOILER CONTROL

1. BOILER CONTROL? ACCOMPLISHED BY PUMP SYSTEM DESIGN
   STEAM TURBINE INTERLOCK? RACER DESIGNED AS FOLLOWING CONTROL
   START UP PROCEDURE? REFER TO RACER CONTROL DATA

2. ICE BREAKING LOAD PROFILE IMPACT ON LIFE? SHOULD BE BENIGN. PLACE RACER IN STANDBY MODE DURING SHIFT.

THE GARRETT CORPORATION
9851-9951 SEPULVEDA BOULEVARD
POST OFFICE BOX 92248
LOS ANGELES, CALIFORNIA 90009
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30 January 1981

Mr. John G. Wainwright
Melville Shipping Ltd.
Gulf Canada Square West
401 - 9th Avenue S.W.
Calgary, Alberta, Canada
T2P 3C5

Dear John:

It was good to see you again on January 15. As I indicated, we have been working with Klockner-Humboldt-Deutz and Blohm and Voss in Germany on a COGAS version of our 990 gas turbine. I just recently received a copy of the English language version of the paper describing the configuration of this power plant. It is enclosed for your information. As you can see, if all the available waste heat is utilized the total package becomes very efficient (85.4). From our discussions it appears that the ice breaking tanker would in fact be able to utilize all the waste heat developed by this unit.

I am also enclosing a paper describing a waste heat driven air compressor. This unit is an existing product developed for another application that could be installed in the exhaust stack of an LM2500. The large amount of air generated could be used for a number of purposes in the ice breaking tanker. I think you would agree that this is far and away the most efficient way to generate large volumes of compressed air.

As I mentioned in our conversations, we would be quite happy to come to Calgary to review the details of either of these systems with you, along with the current status of our RACER program. On the other hand, you might find it more productive to come to our Phoenix plant where we would naturally have more people available to give you the complete story on these units.

I would be interested in your reaction to the two enclosures. If in the near future you might be able to come to Phoenix, or you wish someone to come to Calgary, please let me know or contact Kevin Elmes in our Toronto Office.

Very truly yours,

Rob Chapman

R. M. Chapman
Market Development - Marine Systems

RMC/vbs
Kevin Elmes

DIVISIONS: AIRESEARCH MANUFACTURING CO. OF CALIFORNIA • AIRESEARCH MANUFACTURING CO. OF ARIZONA
AIRESEARCH AVIATION COMPANY • AIR CRUISERS COMPANY • AIRSUPPLY COMPANY • AIRESEARCH INDUSTRIAL DIVISION
GARRETT INDUSTRIAL SUPPLY COMPANY • AIRESEARCH CASTING COMPANY
SUBSIDIARIES: AERO HYDRAULICS, INC. • [illegible] PRODUCTS, INC. • GARRETT MANUFACTURING LIMITED
GARRETT INTERNATIONAL, S. A.

Enclosures

Combined 6.5 MVA Gas-Steam Turbine Set

with Heat Recovery

GARRETT

KHD
Georg Oberländer
Klöckner-Humboldt-Deutz AG

Blohm+Voss
Dr. Günter Siemer
Blohm + Voss AG

Introduction

The critical energy situation calls more and more for an economical conversion of fuel energy into mechanical or electrical energy and useful heat. Normally, shaft power production is given priority, and one should try to utilize as much of the produced heat as possible. Since any transportation of the heat involves high losses, such energy production site should be in close vicinity to the heat consumers.

Large heat quantities are only needed at a limited number of sites, mainly for industrial processes. In the majority of cases only restricted heat quantities are useful for industry and private households in their immediate neighbourhood. For such sites electrical energy and the heat generated thereby would have to be produced in such small units that both kinds of energy may be utilized without incurring long-distance line losses.

Owing to the critical supply situation of primary energy, the former importance of liquid fuel for these applications will change in favour of natural gas and environmental aspects will also be of great importance.

Bearing in mind such phenomena, a concept is analysed in the following which allows the production of electrical energy with max. possible efficiency in such small units that the additional heat may be exploited in the close vicinity.

Combined steam-gas turbine set

Gas turbines as well as diesel engines may be used for the power range of 5 MW being under consideration here. The efficiency of gas turbines in terms of shaft power is lower, but the useful heat in the exhaust gas is higher and the heat losses resulting from the cooling of machine components and oil as well as from surface radiation are lower.

The useful heat in the exhaust gas of gas turbines may be exploited down to 100° C in the case of operation with natural gas, normally for production of steam and hot water. Apart from using the generated steam for heating purposes, it may also serve for admission of a steam turbine, the power of which adds up to the power of the gas turbine, thus increasing the total shaft power. Even when utilizing the steam in such a manner, there still remains useful heat.

In the following, plants for energy supply are described which can be operated with natural gas and offer a high degree of efficiency both with regard to the production of electrical energy and additional heat utilization. Based on gas turbine MI 990 and steam turbine KR 40/50-2, the plants consist of such components that come up to the latest standard of technology and are available on the market.

Gas turbine MI 990

Gas turbine MI 990 (Fig. 1) is of light-weight design; it is based on the experience with aircraft gas turbines and particularly developed for long-term industrial applications. Its weight/power ratio of 0.8 kg/kW is much lower than that of conventional industrial gas turbines; however, it is considerably higher than that of aircraft gas turbines and their versions for industrial applications. Thus, this turbine combines the ample experience and new technologies in air navigation with design features, such as solid construction longlife and low maintenance requirements, rendering it particularly suitable for industrial applications.

It is a two-shaft engine consisting of a gas generator and a free power turbine.

The air is drawn in by the compressor via an annular intake. The gas generator consists of a two-stage centrifugal compressor driven by a two-stage axial turbine. An annular combustion chamber with 18 injection nozzles is arranged around the gas generator turbine. The hot gases are flowing first through the gas generator turbine and then through the three-stage power turbine arranged behind it. The rotor and stator bladings of the gas generator turbine are air-cooled.

The drive shaft of the power turbine is led through the hollow shaft of the gas generator to the air inlet end of the gas turbine. There it is connected with the input shaft of the gearbox via a toothed coupling. The gearbox is equipped with drives for all gas turbine auxiliaries.

The gas turbine MI 990 draws in the ambient air through a ducted inlet system.
In the air intake duct in front of the gas turbine an air inlet filter will be arranged and a silencer may be provided to reduce the high-frequency intake noise. A silencer is also fitted in the exhaust duct. In the case of exhaust heat recovery plants, the silencer may be fitted upstream or downstream the exhaust heat boiler.

The gas turbine MI 990 is started with a pneumatic starter, which may be supplied with compressed air or in the case of operation with natural gas with the gas under supply pressure. It takes 60 seconds until idling speed is reached with liquid fuel and 3 minutes with natural gas.

The gas turbine MI 990 may be operated both with liquid fuels or with natural gas. Dual fuel operation is also possible allowing fuel change during idling or during operation under load.

To lubricate turbine, gearbox and generator a mineral oil with a viscosity of 32 cSt. is used. Lube oil consumption is below 0.4 l/h.

The gas turbine is equipped with an electronic 24 V control system providing for automatic start, run-up to full speed, governing, alarms, safety devices and emergency shutdowns.

The turbines of the 990 series are so designed that they are suitable for boroscope inspections, i.e. they need not be disassembled for such purpose. To simplify repairs the turbine is of modular construction. These modules are: output shaft with main and accessory gearboxes, gas generator, combustion chamber and power turbine.

Thanks to its long service life combined with modular construction the engine normally need not be removed for repairs and overhauls.

As the gas turbine MI 990 is a two-shaft engine, the gas generator adapts itself to the required power with regard to speed, air flow rate, pressure ratio and hot gas temperature. The speed of the power turbine is kept constant for the generator drive. Under ISO reference conditions (0 m altitude, 15° C) the following values apply to full load operation:

air flow rate	20 kg/s
compressor pressure ratio	12
gas generator speed	17 830/min
power turbine speed	7 200 /min

- 5 -

The curve of variable values is shown in fig. 2 considering the intake and exhaust pressure losses. For complete plants it is necessary to consider the losses caused by the gearbox and the silencer in the exhaust-gas system and for the determination of the power at the generator terminals the generator efficiency.

Exhaust-gas boiler

An exhaust-gas boiler is arranged downstream the gas turbine which has been developed for waste-heat recovery from Diesel engines, gas turbines and industrial processing plants.

These exhaust-gas boilers are noted for their high reliability and economy. They require little maintenance and are therefore suitable for watchfree operation.

These features are the result of many decades of experience in boiler construction for marine and industrial use.

Depending on the particular application, the exhaust-gas boilers are built either as natural-circulation types, forced-circulation or once-through types and optimated for the process in which they are to be integrated. The exhaust-gas boiler of the combined units described herein operates on the forced-circulation principle.

The boiler consists of three elements, the two-stage economizer, the evaporator and the superheater. The heating surfaces consist of double-ribbed tubes, constructed as U-shaped tube coils, and are connected with the inlet and outlet headers (Fig. 3). A rugged, vibration-resistant steel frame supports the tube systems as well as the sheetmetal cladding and insulation.

- 6 -

The casing is reinforced by sectional steel and is gastightly welded. It is provided with lateral access openings. The front and the rear of the boiler casing are so designed that the tube coils can be withdrawn.

For cleaning the heating surfaces rotary-tube soot blowers are installed between the tube bundles on fuel oil operation.

The mountings consist of forged steel or cast steel; casing seat and cone are of corrosion-proof material.

The exhaust-gas boilers are effectively heat-insulated by means of mineral wool mats and a galvanized steel sheeting.

Steam turbine

The condensing steam turbine series KR-2 in reaction design with welded casing as package unit was developed in particular for waste-heat recovery on motorships and in industry. The program of manufacture covers powers from 600 kW to 6000 kW.

The required turbine type and size can be selected on the basis of the live steam and exhaust steam flow volumes by means of the sizing diagram (Fig. 4).

Based on this turbine series the optimum type can be selected for the combined gas-steam turbine unit herein described.

The salient features of these condensing steam turbines are the drum-type rotor, the reaction blading, the thermal-elastic design of the inner casing as well as the largely symmetrical top and bottom casing halves.

The high reliability of this turbine is based on the thermal-elastic design of the steam carrying parts.

Fig. 5 shows the double-shell structure of the turbine casing with horizontal joint in the centre-line. In this case the outer casing and the separate bearing pedestals are of welded construction while the inner casing is of cast steel. Due to the design of the outer casing - the inner casing is surrounded by exhaust steam - a bearing span has been obtained which is especially short for a condensing turbine. This measure has a positive effect not only on the total length of the unit but also on the dynamic behaviour of the turbine. The arrangement of the bearing pedestals and the outer and inner casings to each other ensures the safe alignment of outer and inner casings to the rotor and thus the radial blade clearance under all load and temperature conditions.

Further advantages of this double-shell turbine design are that:

- apart from the control valve chest only the inner casing in way of the inlet and the nozzles is subject to exhaust steam conditions. For this reason an insulation of the turbine casing is not necessary.

- The top halves of both outer and inner casing are easily removed so that the rotor is easily accessible for inspections. The bottom casing halves remain in their aligned position on the foundation.

The connection between the inlet chest and the inner casing is formed by the diffusors welded into the chest and projecting into the inner casing.

The stuffing boxes at the casing ends as well as the balance piston are of the labyrinth type.

The turbine rotor which is designed for subcritical running is machined from a solid forging. The journal bearings are of the multi-face sleeve type. The axial forces produced by the pressure difference in the blading are absorbed by the balance piston and a double-sided segmental thrust bearing.

The available heat drop is utilized in a single- or two-row control stage (depending on the heat drop), followed by a multi-stage reaction part.

The reaction blading which was developed in conjunction with the Technical University of Hannover ensures high reliability and economy due to the following features:

- high mechanical and thermal strength in both guide blades and rotor blades to withstand operating stresses, as well as a large sectional modulus to avoid vibration,

- high blading efficiency at both rated and part load as a result of the large, shock-free entry angle range, and

- best sealing of the blade ends against casing and rotor by a special electrolytic method of thinning the blade tips.

The turbine is equipped with nozzle group control with several control valves which are flexibly mounted on a traverse in the inlet chest in accordance with their order of opening. The traverse is guided in the inlet chest by two spindles and is positively controlled via a linkage from a hydraulic power piston.
The advantages of this arrangement are:

- with only two spindles the steam leakage is minimal.

- constant control of the traverse ensures that the valve spindles cannot stick.

- the oil and steam carrying parts are kept separate from each other.

For speed control the turbines are normally equipped with a hydraulic Impeller control. An inlet pressure control maintains the live steam pressure constant.

Total plant

The total plant comprises the gas turbine module with turbine, air intake system, exhaust gas system and auxiliaries and the steam turbine module with turbine, boiler system, condenser and pumps, the combined gearbox for both turbines with overriding clutch for the steam turbine and plant governing and control systems.

The interfaces between gas and steam turbine modules are provided at the output end at the gearbox, namely those for the heat recovery system at the exhaust outlet of the gas turbine and inlet into the boiler system, those for governing and control at the gas and steam turbine system connections with the overall plant control system and for mounting of both turbines, the combined gearbox and the driven generator on the common base frame.

The engine plant is illustrated in Fig. 6. Space requirement and possible arrangement may be taken from Fig. 7 and 8.

The heat flow diagram selected in this case is made up so as to keep the shaft power as high as possible. Modified versions with reduced construction expenditure or increased share of useful heat in the form of steam or hot water to the detriment of shaft power are possible and may be adapted to the individual requirements.

Heat flow diagram

The heat flow diagram (Fig. 9) is applicable to a combined gas-steam turbine plant which allows the optimum utilization of the heat energy contained in the exhaust gas of gas turbines with relatively low construction costs.

When entering the exhaust-gas boiler, the exhaust gas of the gas turbine still has a temperature of 465° C and cools down to 194° C when passing the tube bundles of superheater, evaporator, economizer 1 and 2. If a further exhaust-heat boiler were provided, further energy could be extracted from the exhaust-gas, resulting in temperature level of 100° C.

The steam system consists of the following main components:

- exhaust-gas boiler with 2 economizers, evaporator and superheater

- steam drum

- deaerator

- 2 heat exchangers

- steam turbine with condenser

- pumps for boiler feed, condensate and circulation

The circulation pump transports boiler water from the steam drum via a heat-exchanger through economizer 2 and the evaporator back to the steam drum. Part of the water is evaporated in the exhaust-gas boiler by the heat in the engine exhaust. Separation of steam from water takes place in the steam drum, from where the saturated steam passes via a superheater located in the exhaust-gas boiler. The superheated steam is then led to the steam turbine and produces mechanical energy. The exhaust steam passes to a condenser and is pumped to the deaerator via a heat exchanger. The deaerator also serves as feed water tank, from where the boiler water is pumped to the steam drum via 2 heat exchangers and the economizer 1. Both heat exchangers have the purpose of improving the heat recovery.

Governing and control

The gas turbine is equipped with an electronic governing and control system which supervises the startability, controls the starting process and run-up to full speed, governs the speed during load run, limits max. power and performs normal and emergency stop functions. During the entire operation safety devices monitor the observance of specific limit values for pressure, temperature and speed, give an alarm when these values are approached and actuate the emergency stop upon reaching these values.

Governing is effected by metering the fuel supply by which the power turbine speed is kept constant, taking into account the given droops. If the speed drops on account of increased power take-off on the generator the fuel supply is increased until nominal speed has been reached again. Owing to the larger fuel supply, the temperature in the combustion chamber rises and the gas generator is accelerated; the compressor thus supplies more air with a higher pressure ratio. Analogously, the fuel supply is reduced on decreasing power and also the compressor speed, air flow rate and pressure ratio.

The gas turbine may be loaded up to the set max. power; in the case of overload, the governing system prevents additional fuel supply and the turbine responds with a speed drop of the power turbine and of the connected generator. If the generator is connected to a large mains, it will maintain the frequency, but will not supply more than max. turbine power to the mains.

Following a short starting period of 1 - 3 minutes, the gas turbine is in a position to supply its full power to the generator and thus to the mains. The steam turbine is now waiting for steam to be produced by the heat of the gas turbine delivered to the boiler since starting. After the min. pressure and temperature values have been reached, the steam turbine is admitted and also delivers its power to the generator. This warming-up process takes 20 - 30 minutes and during said period the generator has to be driven by the gas turbine alone.

In order to prevent that the steam turbine is driven by the gas turbine during this time, an overriding clutch is provided at its input drive to the gearbox.

A similar effect occurs - although to a much lesser degree - on each load increase when the gas turbine responds immediately, the steam turbine, however, only with a delay of several seconds or minutes.

Vice versa, upon each load drop the gas turbine must first reduce its power by the total load difference, whereas the steam turbine power subsequently drops with delay. Due to the fact, however, that the gas turbine power considerably exceeds that of the steam turbine, accounting for approx. 75 %, the recovery performance is not substantially affected in the case of step-load changes.

For this plant the steam turbine is so governed that the generated steam is optimally converted internally into mechanical energy independent of the governing system for the overall plant. The steam turbine is therefore provided with a pilot pressure control which keeps the steam pressure in front of the control valves constant by changing the control valve stroke.

- 13 -

The sudden load rejection with subsequent emergency stop has to be dealt with separately.

In such a case the fuel supply to the gas turbine is cut off so abruptly by a quick-closing valve that the permissible speed limit is not exceeded even when shutting down the turbine from full load. The gas turbine cannot act as brake, however, and absorb the full power of the steam turbine which also in this case would only drop with delay.

In order to avoid non-permissible overspeed, a quick-closing valve is actuated for the steam and a bypass line around the boiler is opened for the exhaust gas so that the heat supply to the boiler is interrupted.

This bypass line remains also open when for special reasons, such as maintenance work, the gas turbine is to operate without the steam turbine.

Power limits

The power output of the gas turbine is dependent on ambient conditions, i.e. temperature and pressure of the drawn-in air. It decreases with rising temperature and falling pressure. The governing and control system ensures that those parts of the turbine blading which are stressed most remain within permissible speed and temperature limits under any ambient conditions. In this case it is distinguished between continuous power for long-term duty and max. power for short-term speak-load and emergency operation.

The power of the steam turbine only depends on the steam production and thus on the exhaust volume and temperature of the gas turbine. As the power of the steam turbine amounts to approx. 25% only, the power limiting characteristics of the overall plant are similar to those of pure gas turbine operation (Fig. 10).

Energy balance

The plant herein described may be reasonably applied where both the additionally produced steam and the water heated up by the condenser are required throughout the year. Heat accumulators may also be used for lopping daily or process peaks.

As fired heating boilers convert the fuel energy into heat with a similar efficiency as the total plant, any extra requirements should be covered by such boilers, particularly seasonal or daily extra demands.

In order to ensure max. utilization of the heat additionally generated, it is essential that the appropriate site be selected, bearing in mind the very high heat losses of approx. 30 % over a 2 km line.

The heat flow diagram (Fig. 11) shows utilization and losses of the fuel energy. It makes clear that the available shaft power already reaches 40 % and allows a total utilization up to 85 %. Nearly half of the utilized energy is converted into mechanical energy.
Furthermore, it becomes quite clear that the residual heat in the exhaust gas is quite low with 12 %.

The total efficiency curve plotted against shaft power is shown in Fig. 12, which has been prepared for steady-state operation.

Summary

The above concept of a combined steam-gas turbine set with heat recovery fits into a power class which makes it possible to install such plant in the close vicinity to consumer centres where both the generated electrical energy and the additional heat may be fully utilized without incurring any long-distance line losses. In most cases it will be necessary to install several plants for such decentralized energy supply and by cutting off part of the plants during times of low energy demands it will be possible to have the remaining units running nearly at full load, that is with optimum efficiency.

Another alternative would be to combine two gas turbines with one steam turbine for double power output.

For applications with large demands for high-temperature heat the total plant may also be equipped with back-pressure instead of a condensing turbine. Total plant efficiency is not affected thereby; the percentage of available shaft power decreases in relation to the temperature level. The layout of the steam circuit remains identical except for the condenser which is omitted.

Thanks to its compact design and low emission values the plant described above allows a favourable site selection and energy production and supply to the consumers with such efficienies that can stand comparison with large most modern power stations.

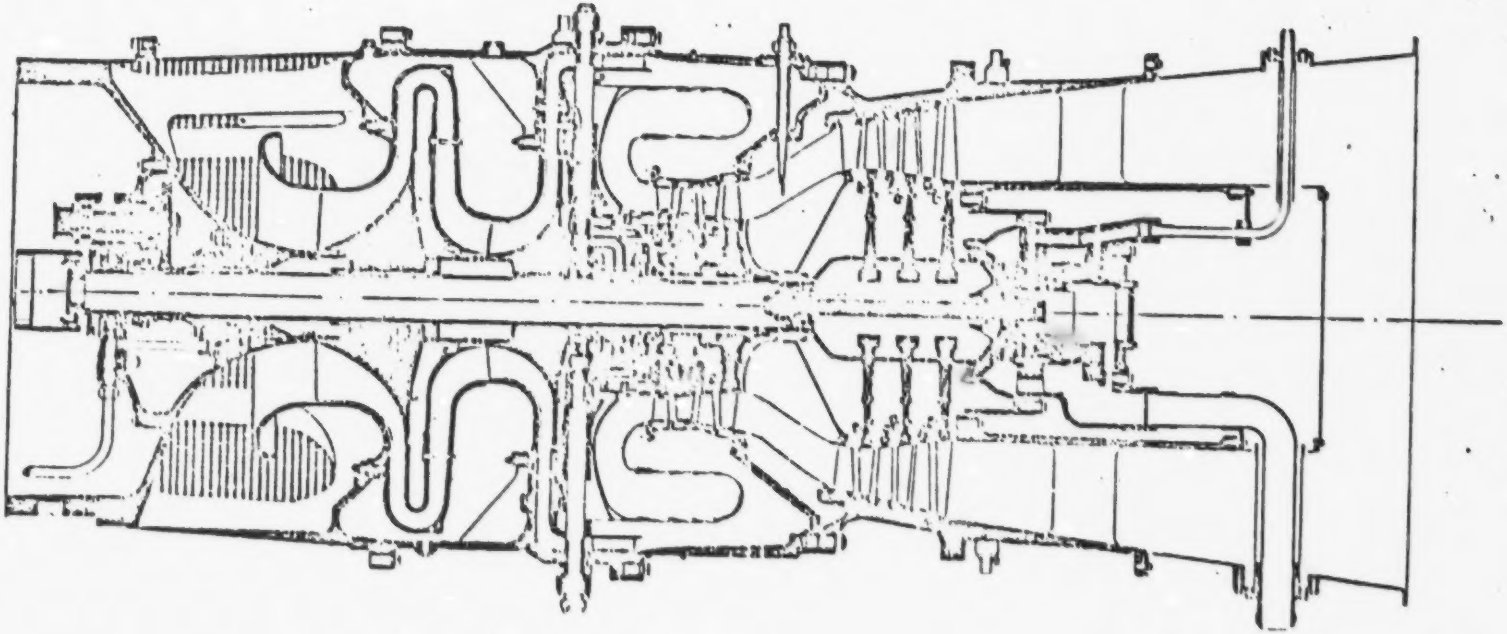


FIG. 1

GAS TURBINE M1990

Blohm+Voss

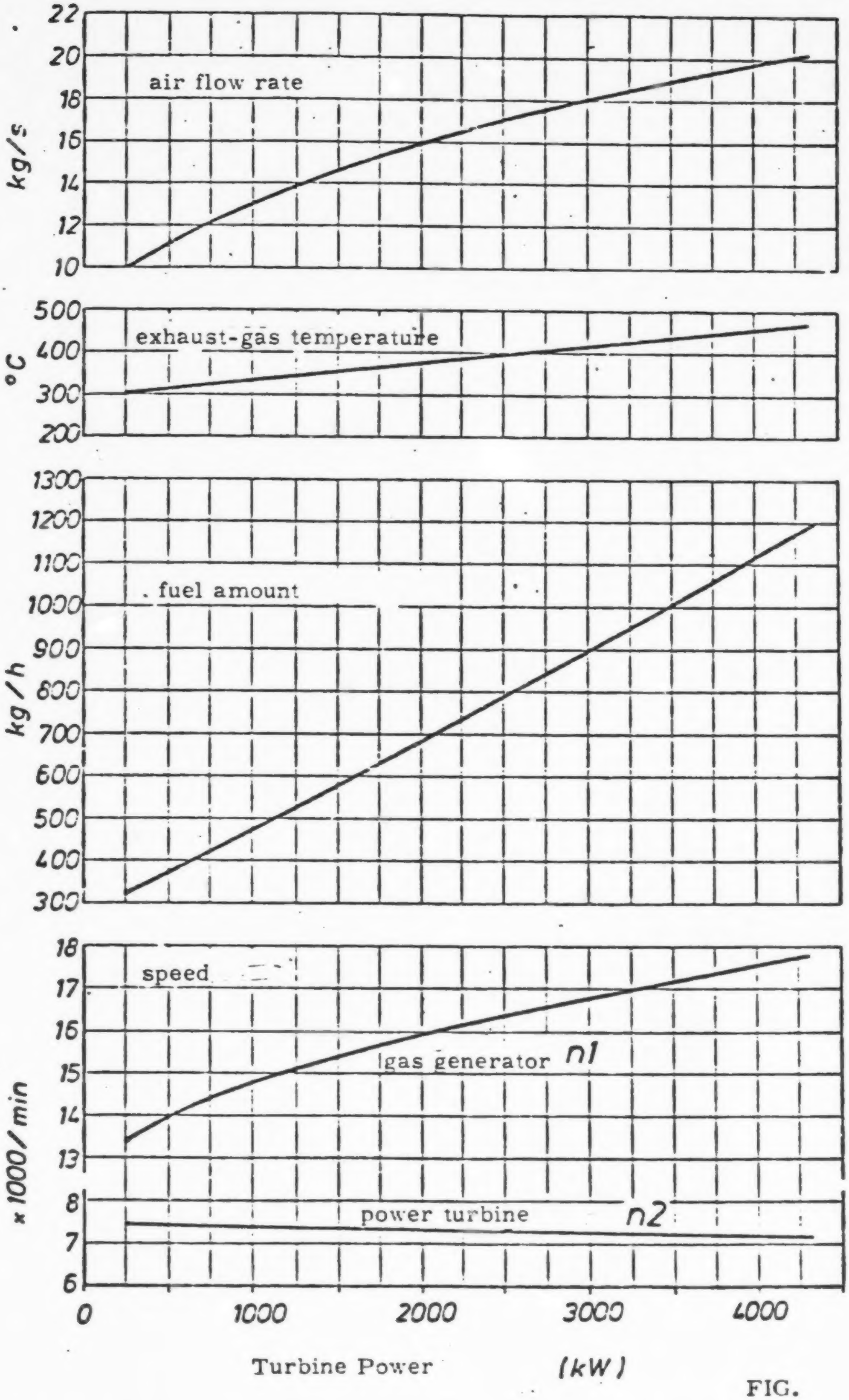


FIG. 2

KHD DEUTZ

Characteristic Curves
of
Gas Turbine MI 990

Blohm+Voss

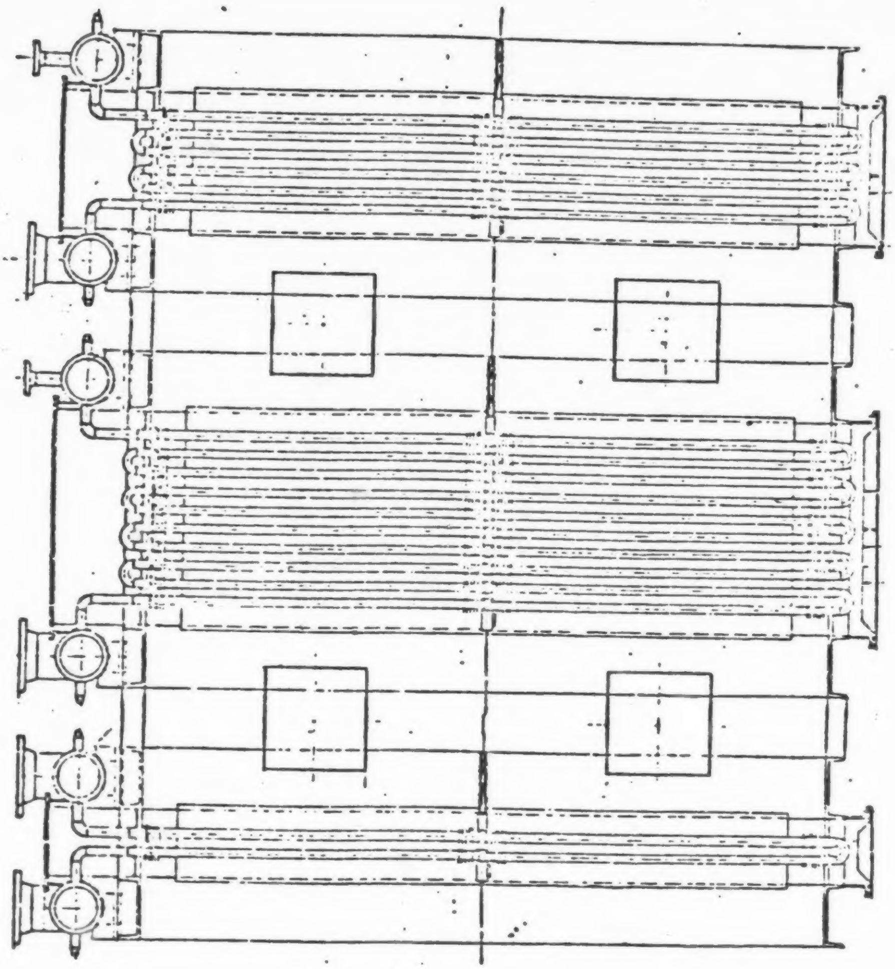


FIG. 3

Exhaust-Gas Boiler

KHD DEUTZ

Blohm+Voss

Condensing Steam Turbine in Reaction Design with Welded Casing

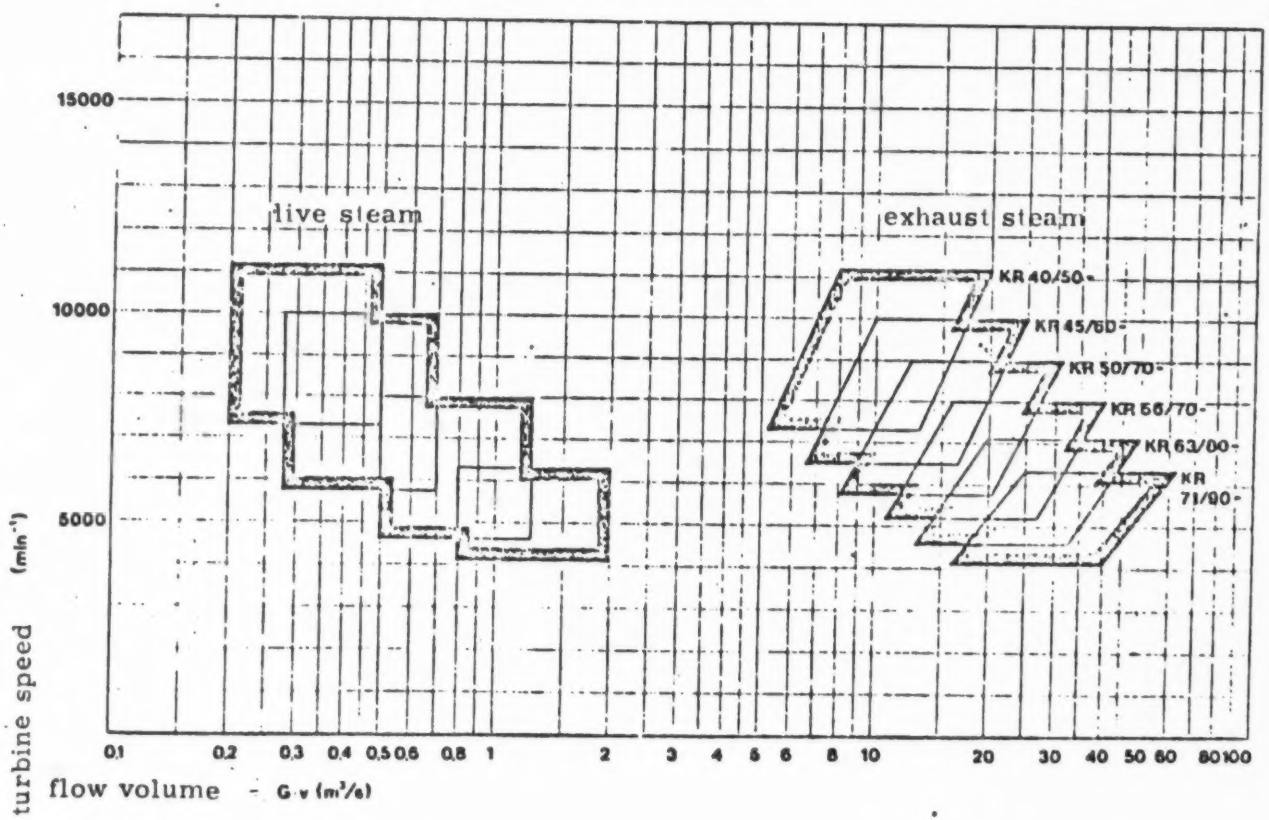


FIG. 4

DEUTZ	Sizing Diagram of Condensing Steam Turbine Series	Blohm+Voss

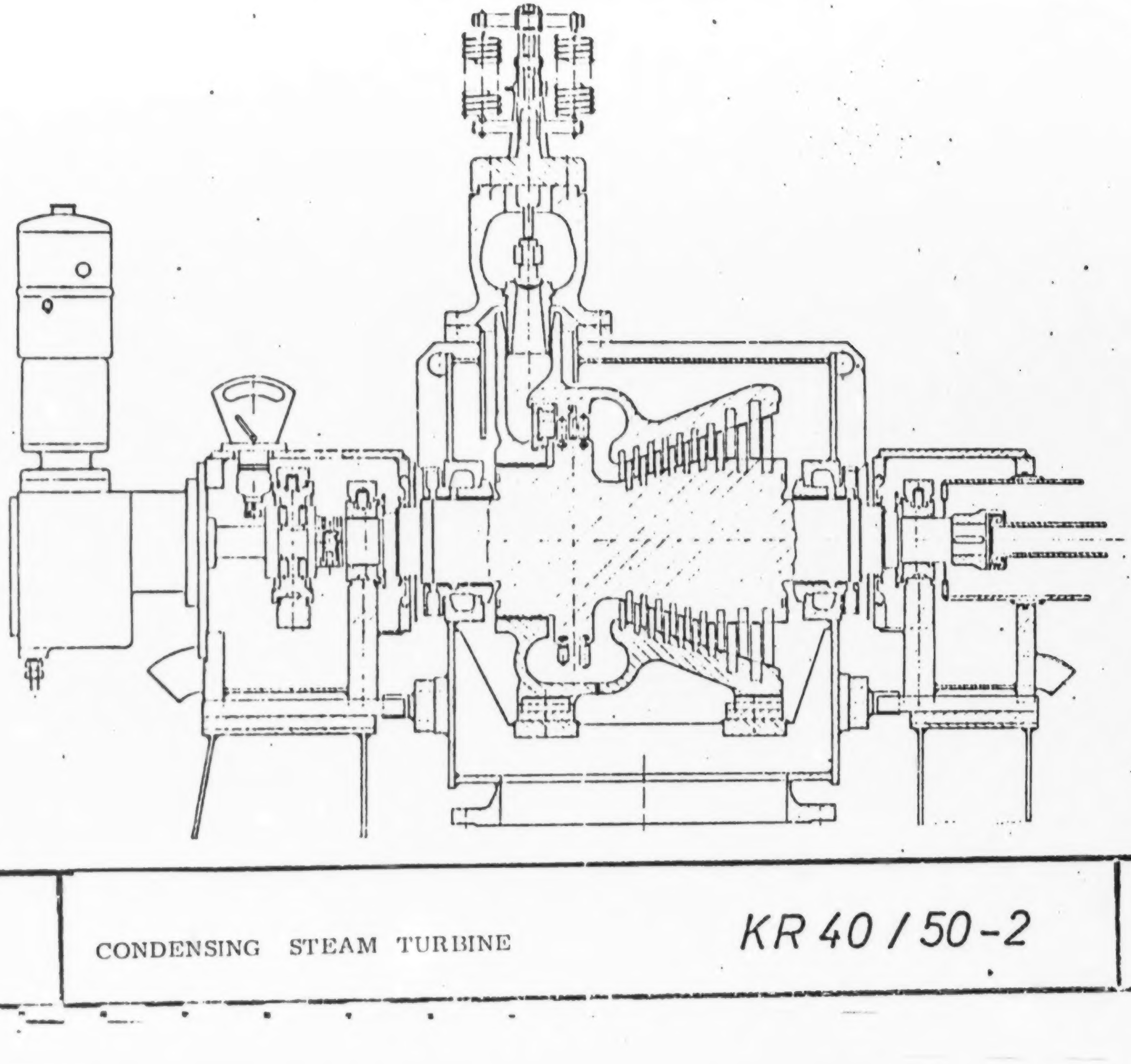


FIG. 5

DEUTZ

CONDENSING STEAM TURBINE

KR 40 / 50-2

Blohm+Voss

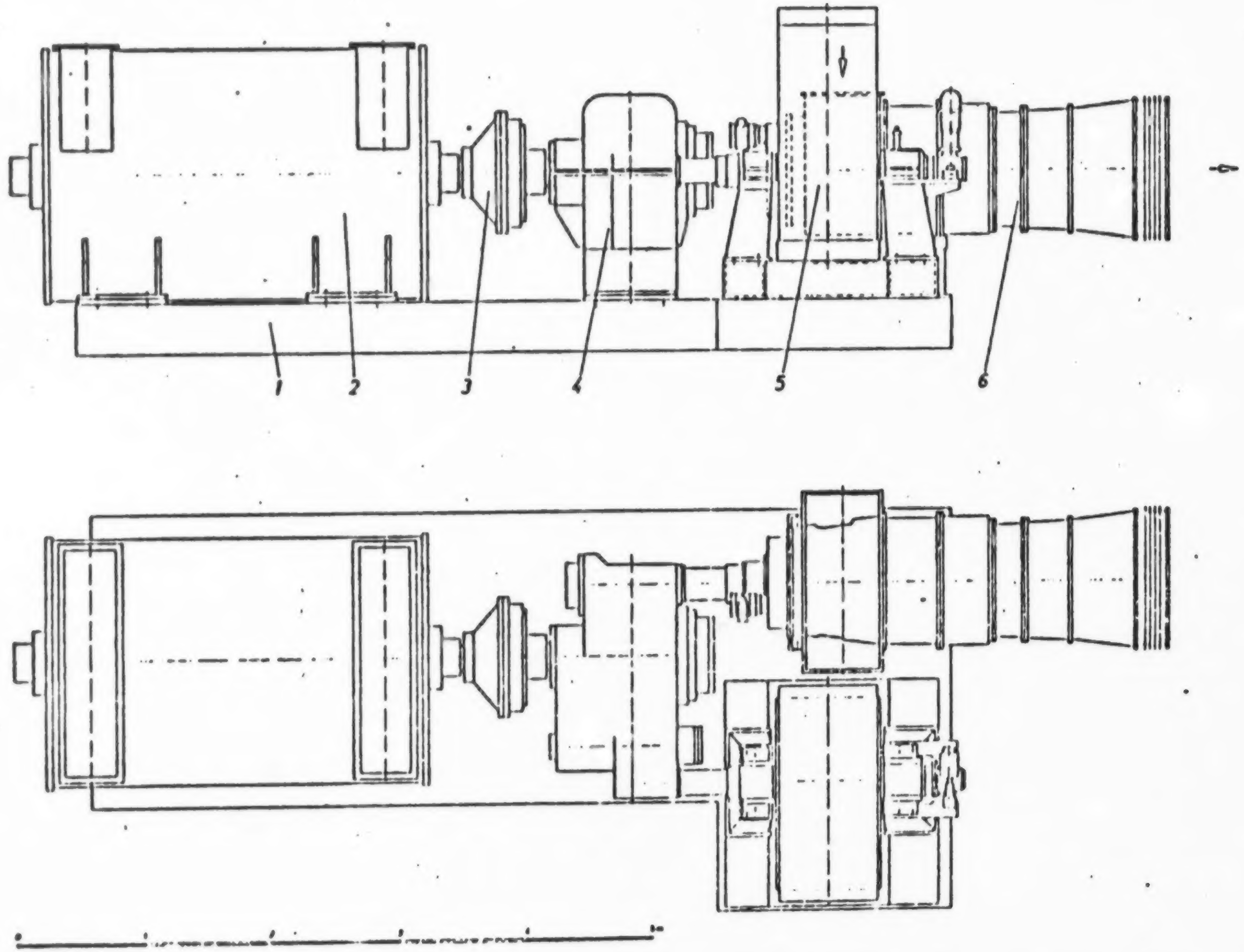


FIG. 6

DEUTZ

GAS-STEAM TURBINE SET

Blohm+Voss

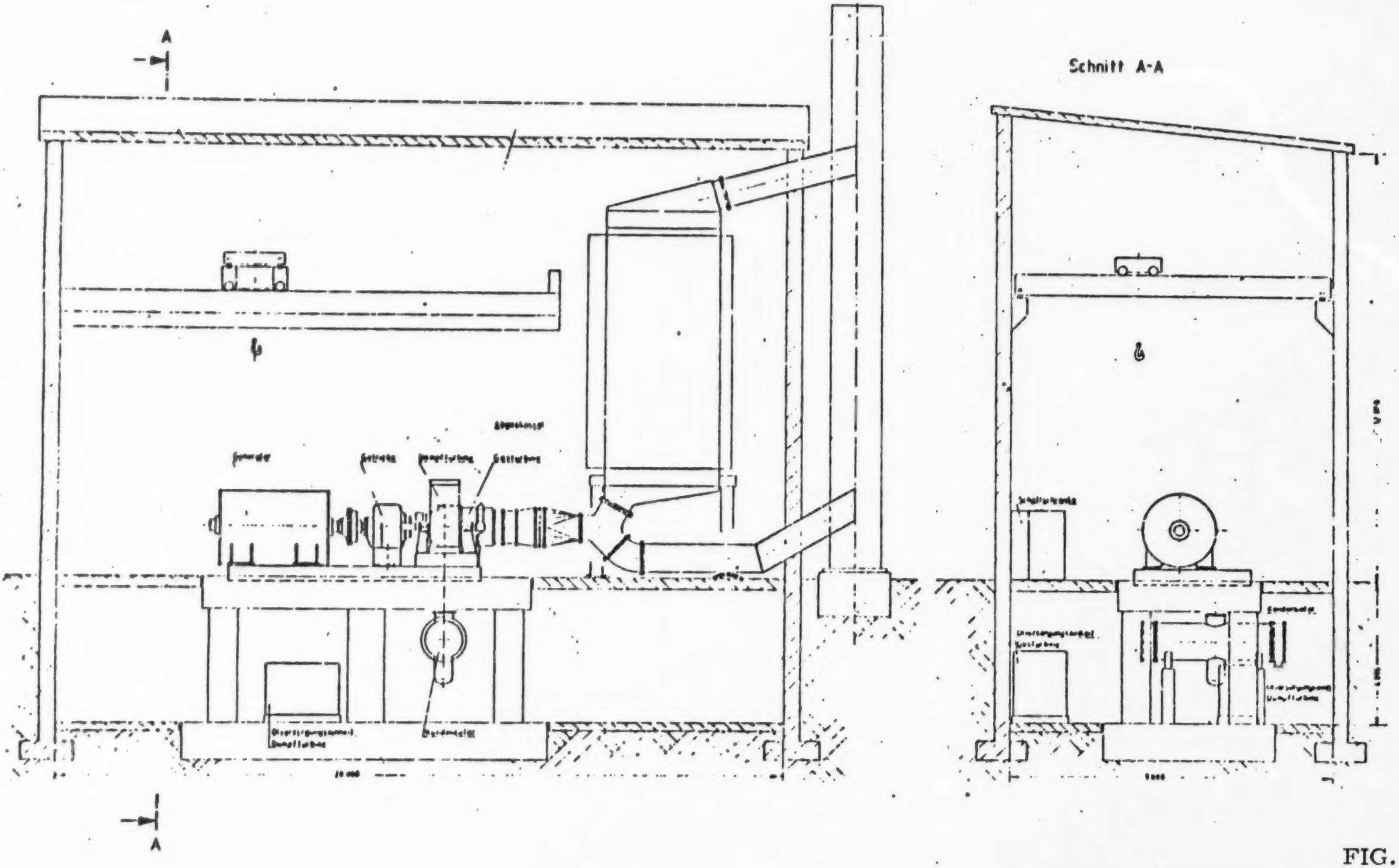

A
A
Schnitt A-A

FIG. 7

ARRANGEMENT PLAN

DEUTZ

Blohm+Voss

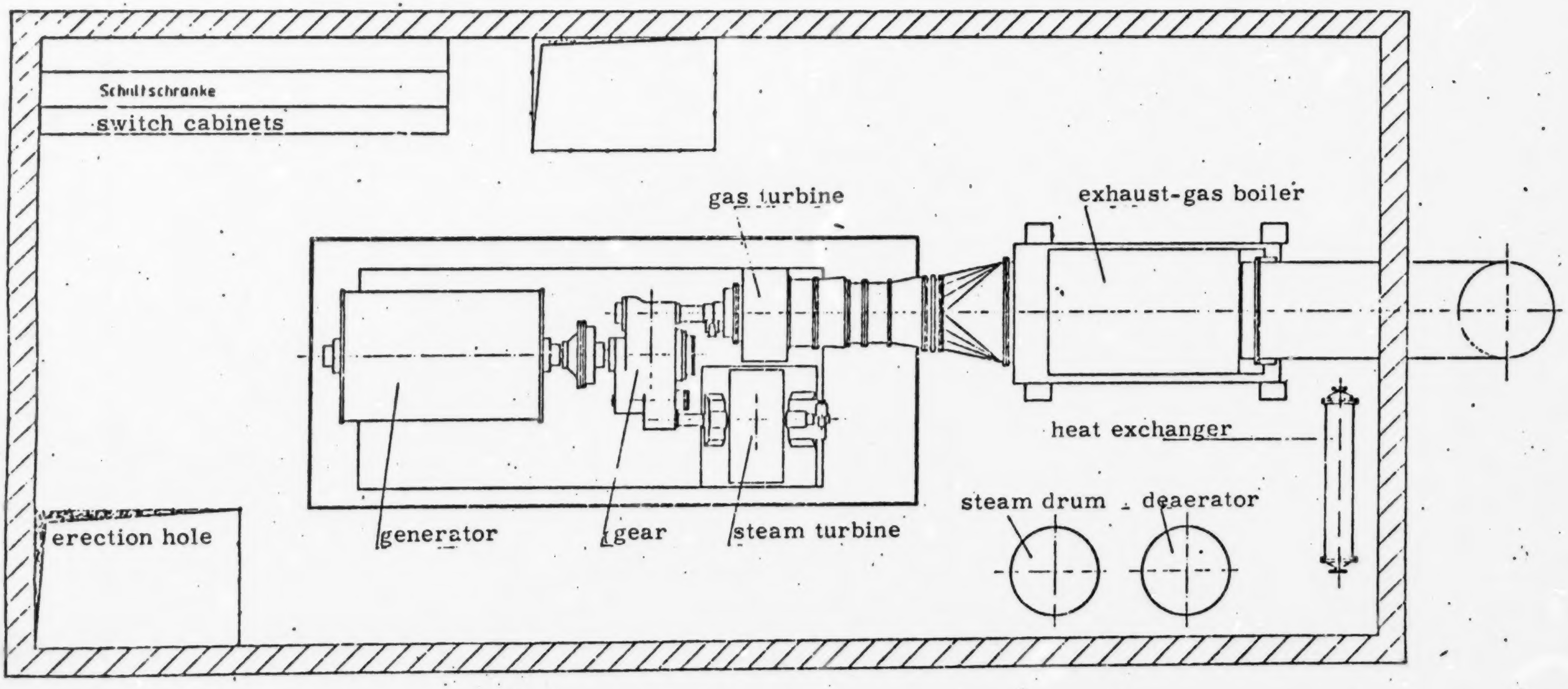

Schaltschranke
switch cabinets
gas turbine
exhaust-gas boiler
heat exchanger
steam drum
deaerator
erection hole
generator
gear
steam turbine

FIG. 8

DEUTZ

ARRANGEMENT PLAN

Blohm+Voss

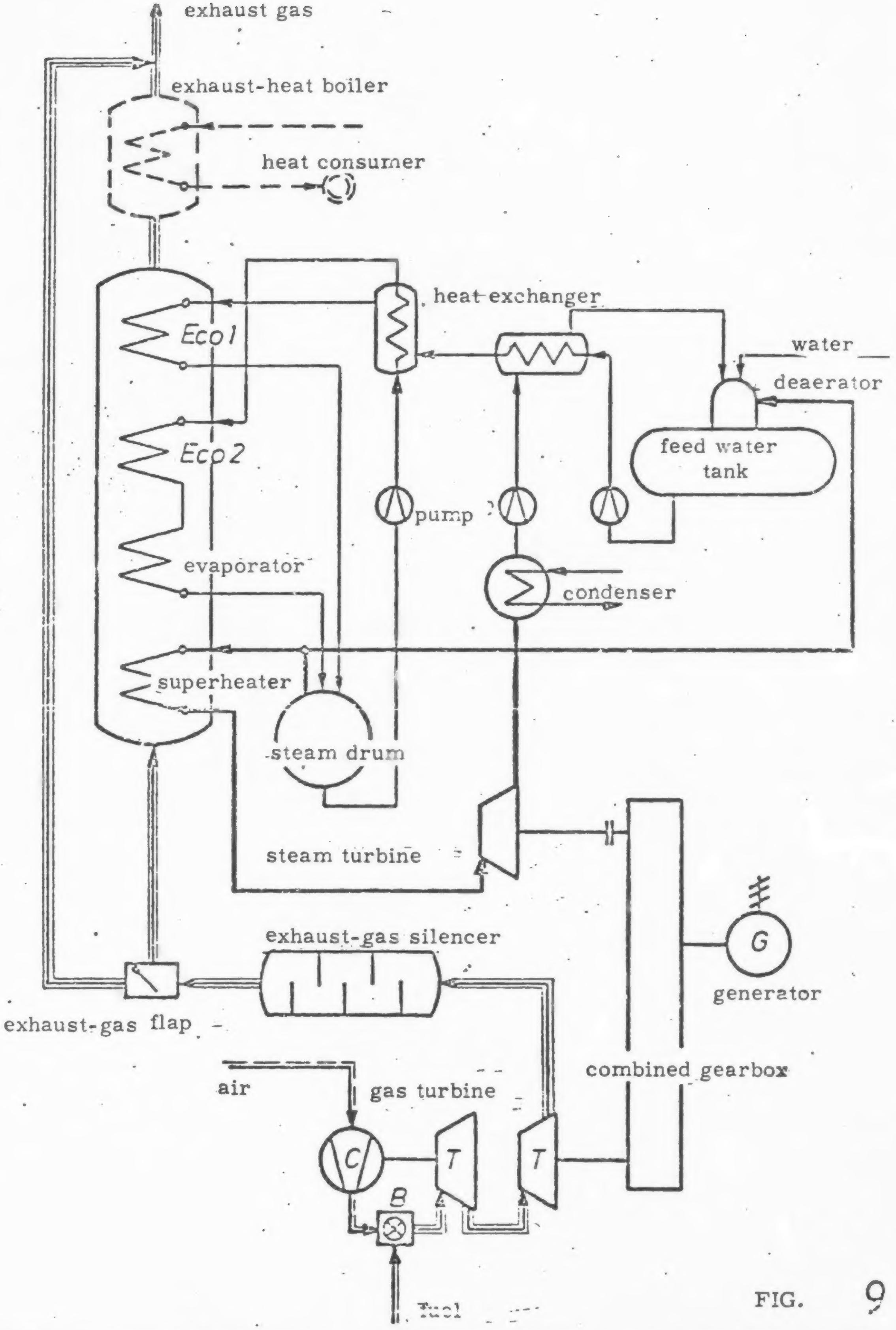

exhaust gas
exhaust-heat boiler
heat consumer
heat exchanger
Eco 1
Eco 2
water
deaerator
feed water tank
pump
evaporator
condenser
superheater
steam drum
steam turbine
exhaust-gas silencer
G
generator
exhaust-gas flap
combined gearbox
air
gas turbine
C
T
T
B
fuel

FIG. 9

KHD DEUTZ

HEAT FLOW DIAGRAM

Blohm+Voss

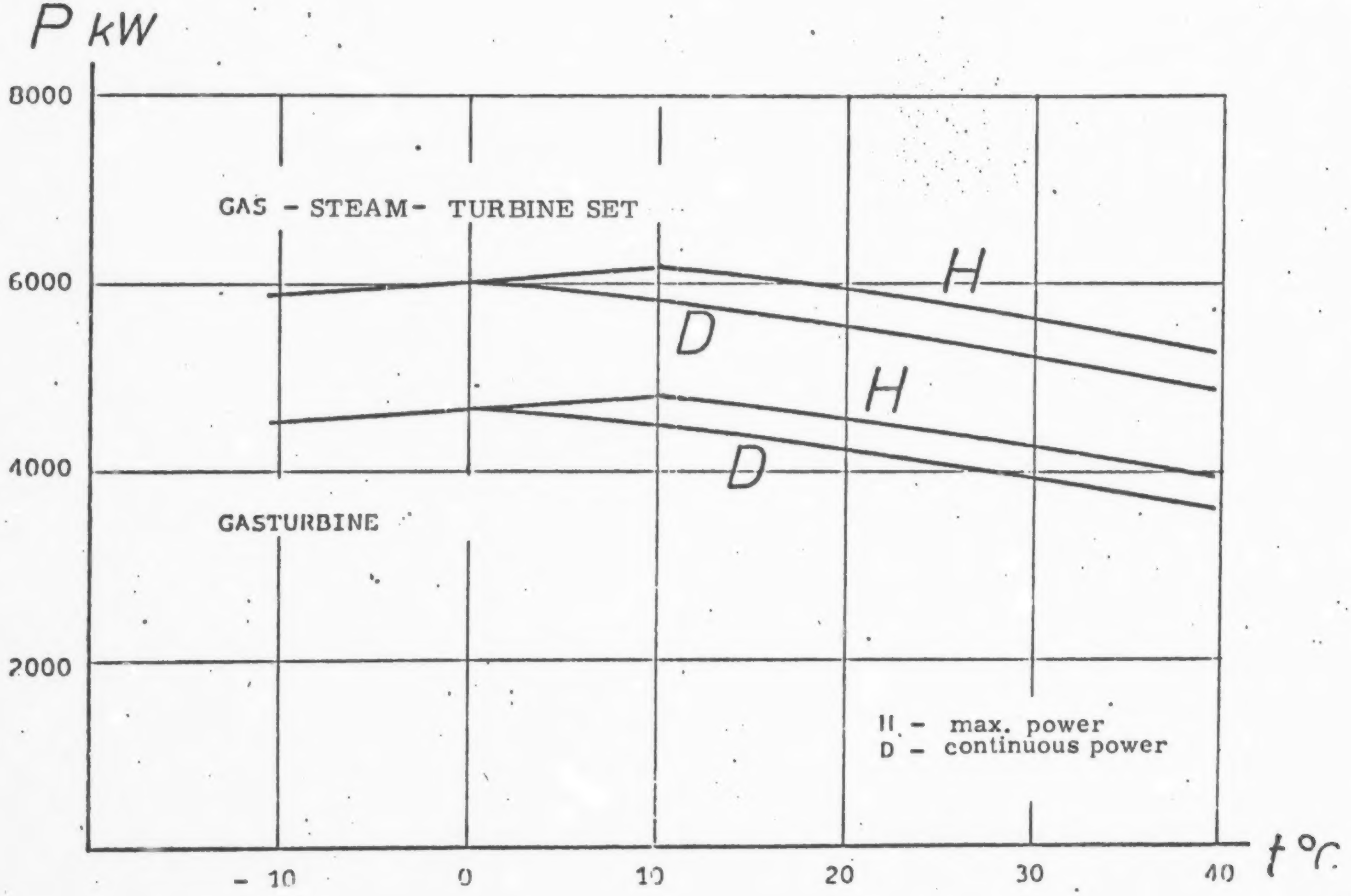
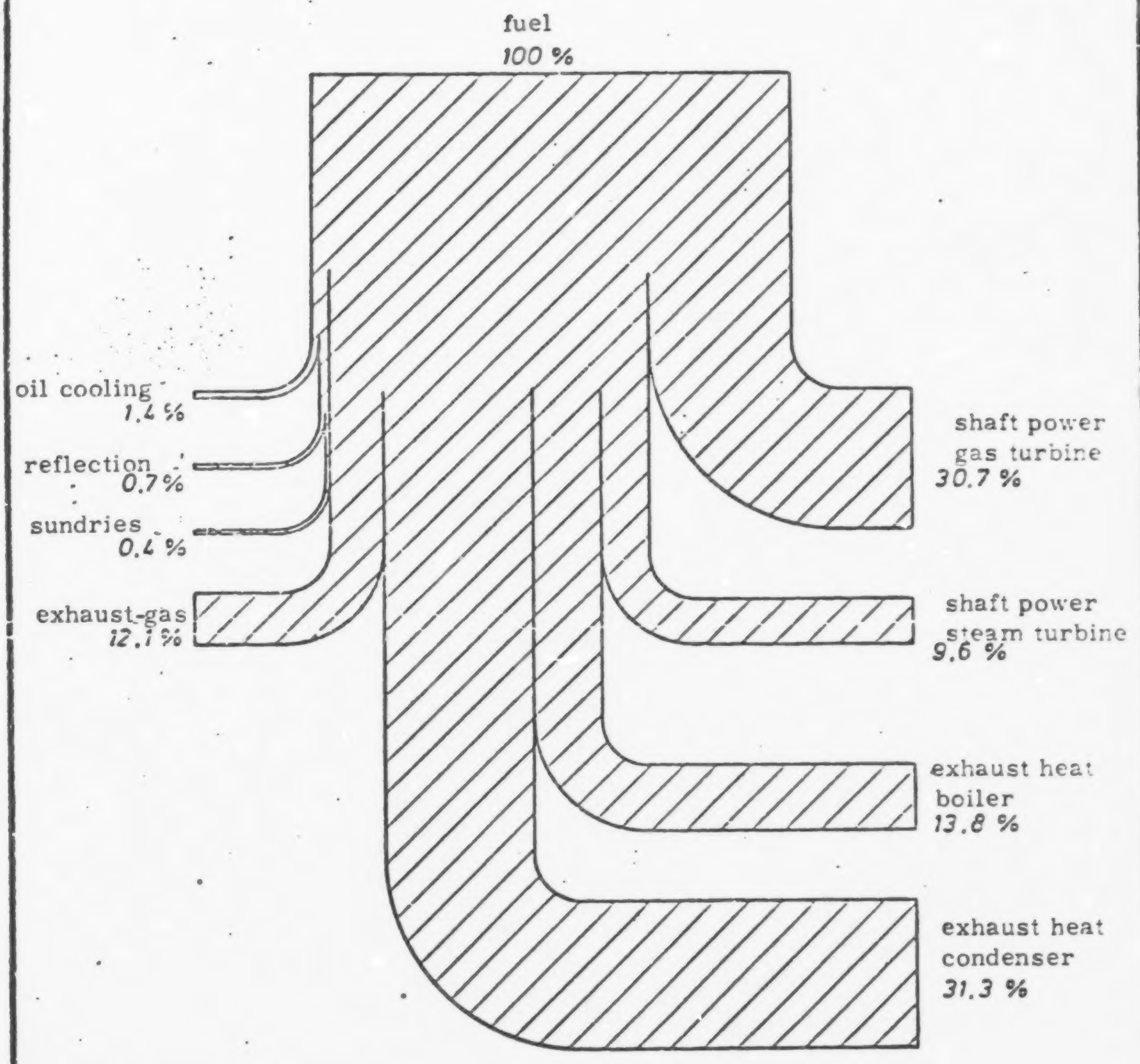


FIG. 10

POWER of GAS - STEAM TURBINE SET

DEUTZ

Blohm+Voss


fuel
100 %
oil cooling
1.4 %
reflection
0.7 %
sundries
0.4 %
exhaust-gas
12.1 %
shaft power
gas turbine
30.7 %
shaft power
steam turbine
9.6 %
exhaust heat
boiler
13.8 %
exhaust heat
condenser
31.3 %

losses

14.6 %

Utilization

85.4 %

FIG. 11

KHD DEUTZ

ENERGY FLOW DIAGRAM

Blohm+Voss

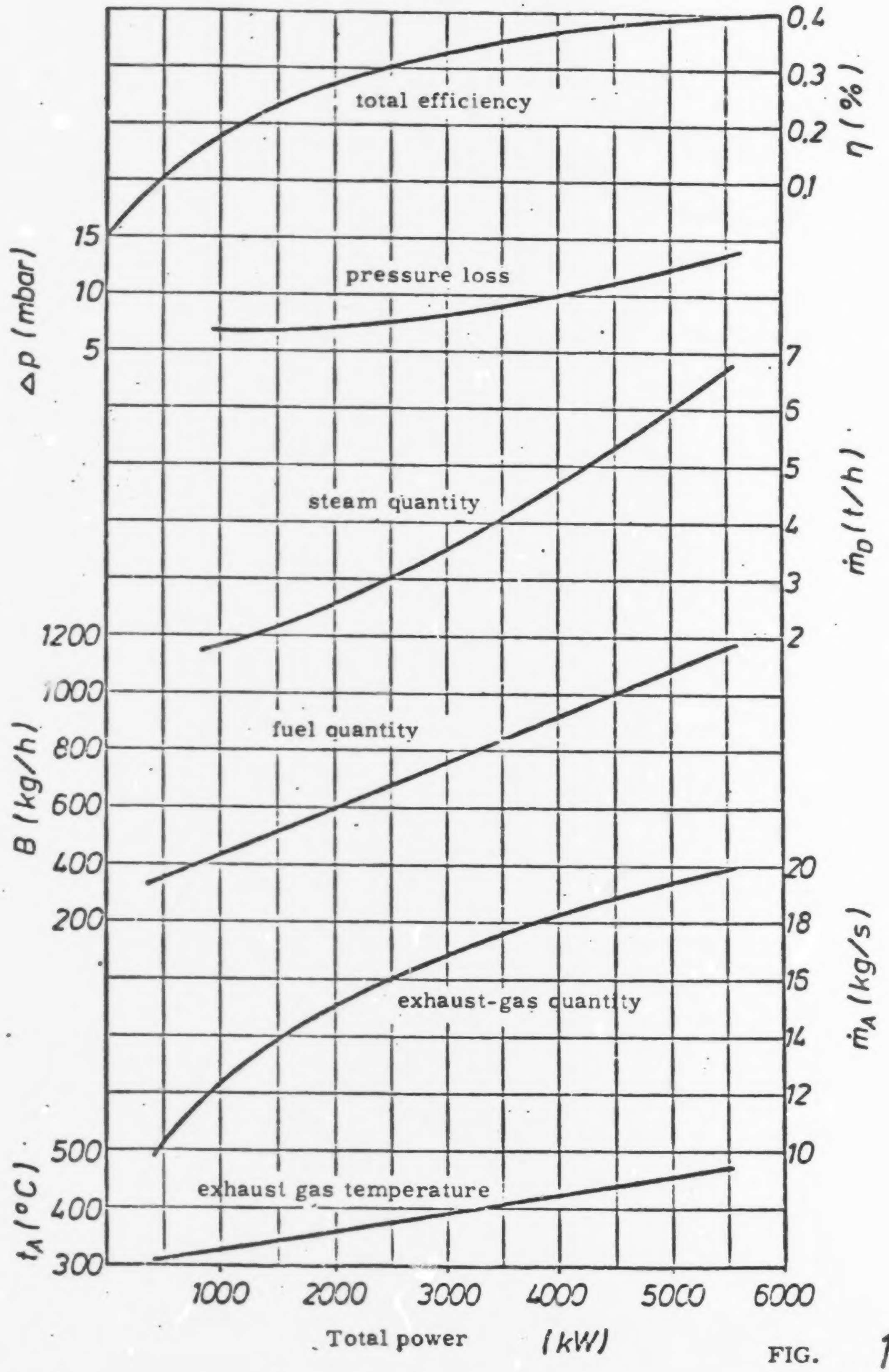


FIG. 12

KHD DEUTZ

Characteristic Curves of
GAS-STEAM TURBINE SET

Blohm+Voss

WASTE-HEAT-DRIVEN AIR COMPRESSION SYSTEM

INTRODUCTION

The air compressor system described below produces 2000 SCFM of air at a pressure of 35 psig. It is powered by the exhaust heat from a single, LM2500 propulsion engine and is activated anytime the engine exhaust temperature exceeds 940°F, or approximately one-half rated power. This concept is depicted in Figure 1. Air is compressed in a compressor to the desired operational pressure. Part of this air is used as the compressed air supply, and the remainder is heated in the heat exchanger and then expanded through a turbine to drive the compressor. The proposed system, shown schematically in Figure 2, consists of four turbocompressors, two engine exhaust heat exchangers, and miscellaneous valves. This system is completely self-contained and requires no support equipment for operation, monitoring, or fault isolation. The advantages of the system are:

- Uses existing AiResearch technology and equipment; therefore, no development program is required.
- Propulsion engine exhaust heat is used to provide the compression energy. Thus, no additional fuel and electrical generation equipment is required. For instance, if this system is used in place of existing systems for 2000 hr per year, the yearly fuel savings for each destroyer is approximately 140,000 gallons.
- Complete redundancy and fault isolation eliminates the need for support equipment.
- The turbomachinery uses AiResearch compliant foil air bearings, which eliminate the need for lubrication and fire protection systems.

Each turbocompressor assembly provides one-third of the required capacity and is independently operated and controlled. Any three of the assemblies can be selected to provide the compressed air. The fourth is used for backup.

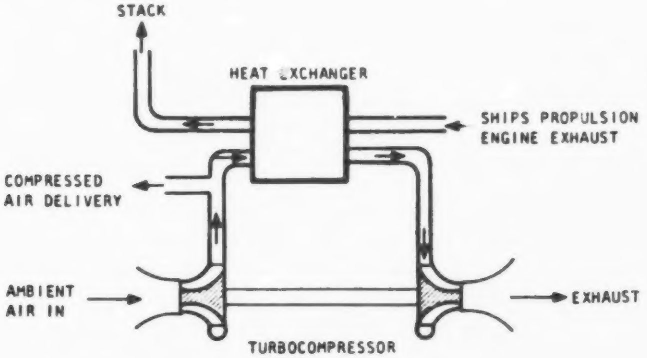


Figure 1. Air Compression System Schematic

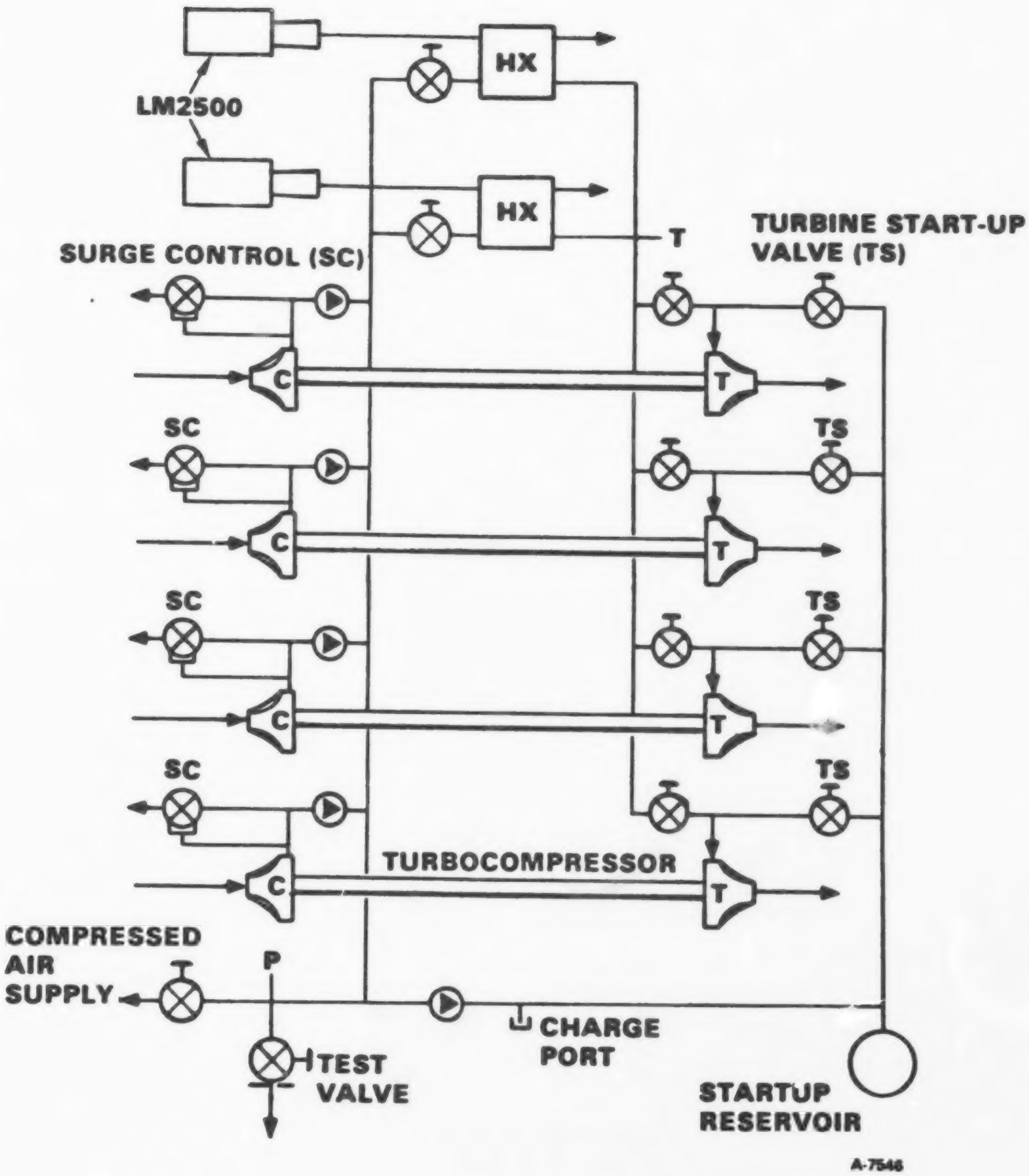

LM2500
HX
HX
T
SURGE CONTROL (SC)
TURBINE START-UP VALVE (TS)
C
T
SC
TS
C
T
SC
TS
C
T
SC
TS
TURBOCOMPRESSOR
C
T
COMPRESSED AIR SUPPLY
P
TEST VALVE
CHARGE PORT
STARTUP RESERVOIR
A-7546

Figure 2. Air Compressor System Schematic

OPERATION

The system is started by closing the turbine inlet isolation valve and opening the turbine startup valve. Compressed air from the startup reservoir flows through the turbine to provide the startup power. When the operating pressure is obtained, as measured on the system pressure gauge, the turbine inlet isolation valve is opened to allow the compressor discharge air to flow through the engine exhaust heat exchanger where it is heated to the design operating temperature before it is expanded through the turbine. The turbine startup valve is closed, and the startup reservoir is filled by the excess compressor capacity. The second and third systems are started in the same manner. When these systems are operating, the compressor air supply valve is opened to provide air to the compressed air delivery system. The fourth assembly is started if system pressure cannot be maintained. Fault isolation and repair can therefore be delayed until system operation is not required.

The major component in the system is the turbocompressor. The proposed unit is the full-flow, fuel cell turbocharger developed by AiResearch for the Power Systems Division (PSD) of United Technologies. These air bearing units were built in two sizes and delivered to PSD under contract from a consortium of nine major U.S. utility companies and the Department of Energy. In that program, the turbochargers are used to pressurize the fuel cell and recover system waste heat.

Figure 3 shows the cross section and a photograph of the full-flow turbocompressor. Important turbocompressor parameters are listed below.

	Partial-Flow Unit	Full-Flow Unit
Speed, rpm	35,000	23,000
Flow rate, lb/sec	4.4	12.8
Power, hp	435	1220
Rotor weight, lb	39	120
Turbine efficiency, percent	90.0	91.0
Compressor efficiency, percent	79.0	82.5
Overall efficiency, percent	69.0	73.0

The other major component, the exhaust gas heat exchanger, also uses existing AiResearch technology. Compact plate-fins initially developed for aerospace applications have been adapted to industrial gas turbine regenerators. This plate-fin matrix provides a compact, lightweight heat exchanger that greatly reduces installation problems onboard ship. Existing plate-fin designs will be combined in the manner that best fits the space available for installation.

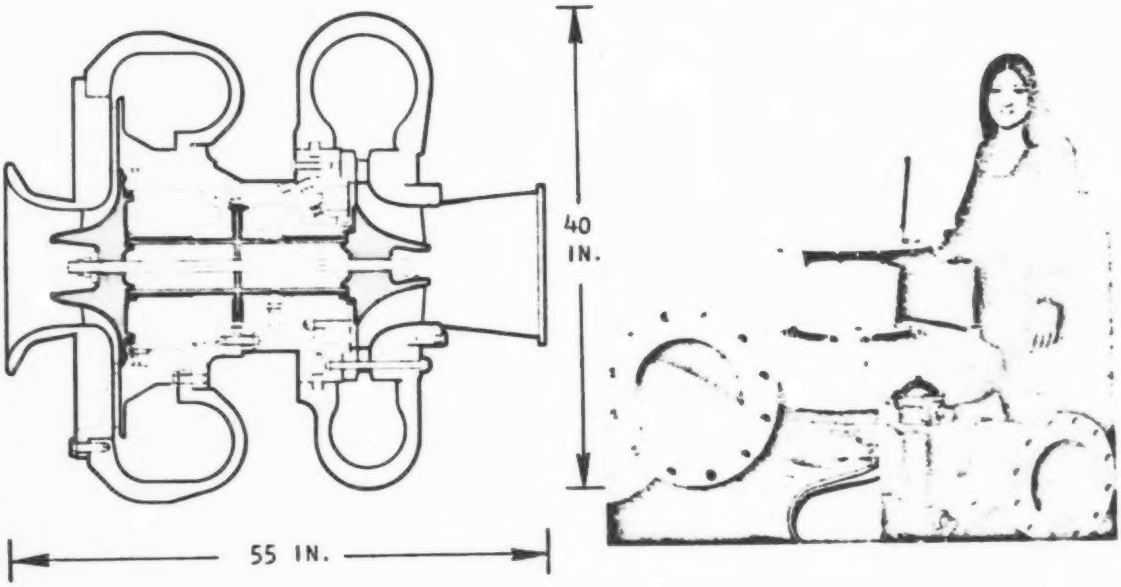
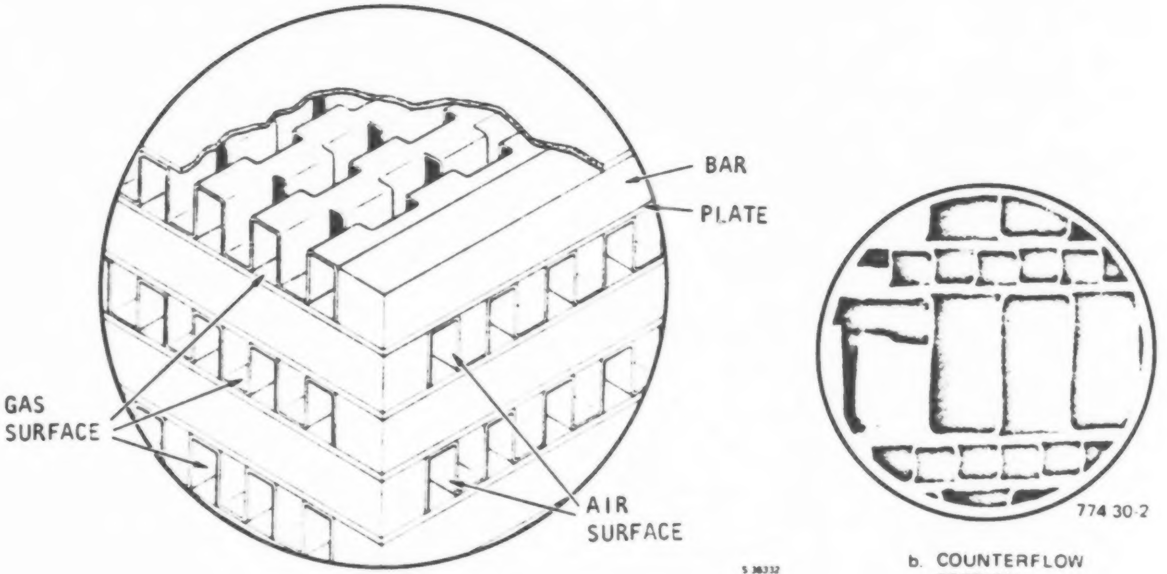


Figure 3. Full-Flow Turbocompressor

F-32906

The plate-fin matrix (Figure 4) consists of layers of corrugated sheet stock (fins) separated by plates. The fins, plates, and edge bars are stacked in a fixture and brazed to form an integrated core assembly. Alternate passages formed by the plates and bars are allocated to each flow stream, and the flow configuration is determined by the arrangement of fins, bars, and welded manifolds. Figure 4a shows a simple crossflow arrangement. Counter, parallel, and multipass flow configurations also are feasible. Figure 4b shows the stacked fin layers in a heat exchanger counterflow section.

The remaining items are all commercially available. Therefore, this high-technology system can be supplied in a short schedule with little risk because no development program is required.


BAR
PLATE
GAS SURFACE
AIR SURFACE
774 30-2

a. END SECTION

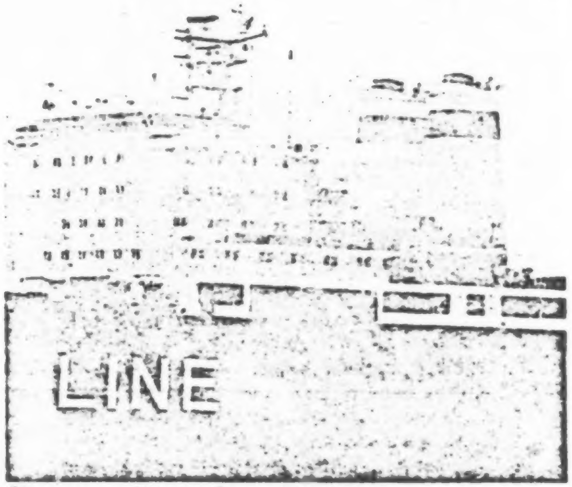
b. COUNTERFLOW SECTION

F-31305 A

Figure 4. Plate-fin Heat Exchanger Construction

Russian COGAS in the Thames

A surprise arrival at Tilbury recently on the homeward container run from the US Gulf was the Soviet merchant fleet's flagship, "Kapitan Smirnov", first of two 25-knot Ro-Ro vessels from the Nikolaev Black Sea shipyard. The vessel's combined cycle machinery applies waste heat from the gas turbines to the production of steam used in a bottoming cycle for supplementary propulsion gas turbines



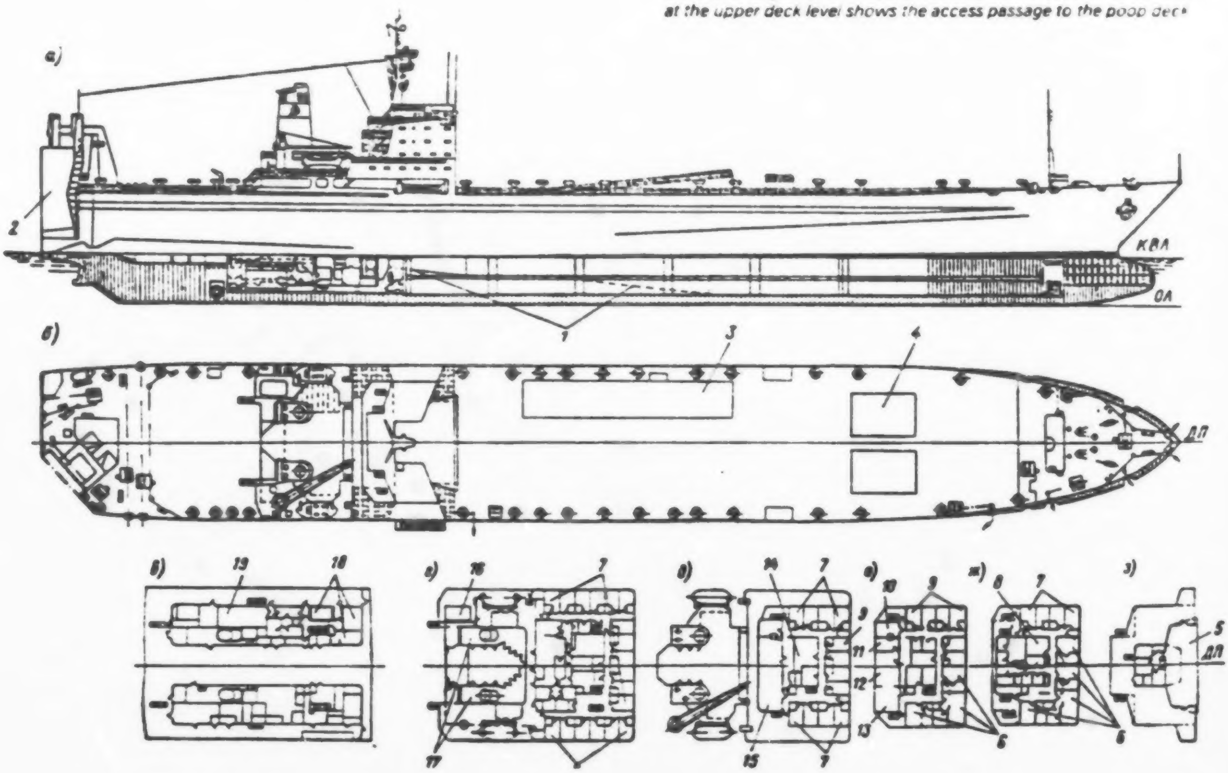
Close-up view of 'Kapitan Smirnov' clearly showing the large diameter exhaust stacks. Combustion air is drawn in through a filter house between the funnels. Note the roadway under the superstructure (Laurence Dunn)

It is ironic that when most of the gas turbine-powered merchant vessels in the world are due for conversion to diesel propulsion, or firm plans are well advanced for such conversion, the Soviet Union has commissioned a large and fast Ro-Ro ship, the *Kapitan Smirnov*, propelled by machinery of broadly similar type, but with significant differences. The ship was delivered some time ago and has made a voyage to the Far East. She is now engaged on the Balt Gulf Line sailing between the West Indies and North and South American mainland Gulf ports to the UK, Continent and Leningrad.

Somewhat smaller than the *Parallas* built by Eriksbergs in the early part of the last decade and upon which the design has clearly been based, the hull form is conventional with a bulbous bow and full shaft bossings (a semi-catermaran afterbody was considered), two thrusters and a semi-balanced rudder. A double skin extends up to the main deck in way of the cargo spaces to form wing ballast and trimming tanks.

Access is over a MacGregor 38m long by 7m roadway quarter ramp capable of carrying loads of 65tons. This is set at an angle of 40degrees to

General arrangement of 'Kapitan Smirnov'. The superstructure plan at the upper deck level shows the access passage to the poop deck



BALT-GULF LINE

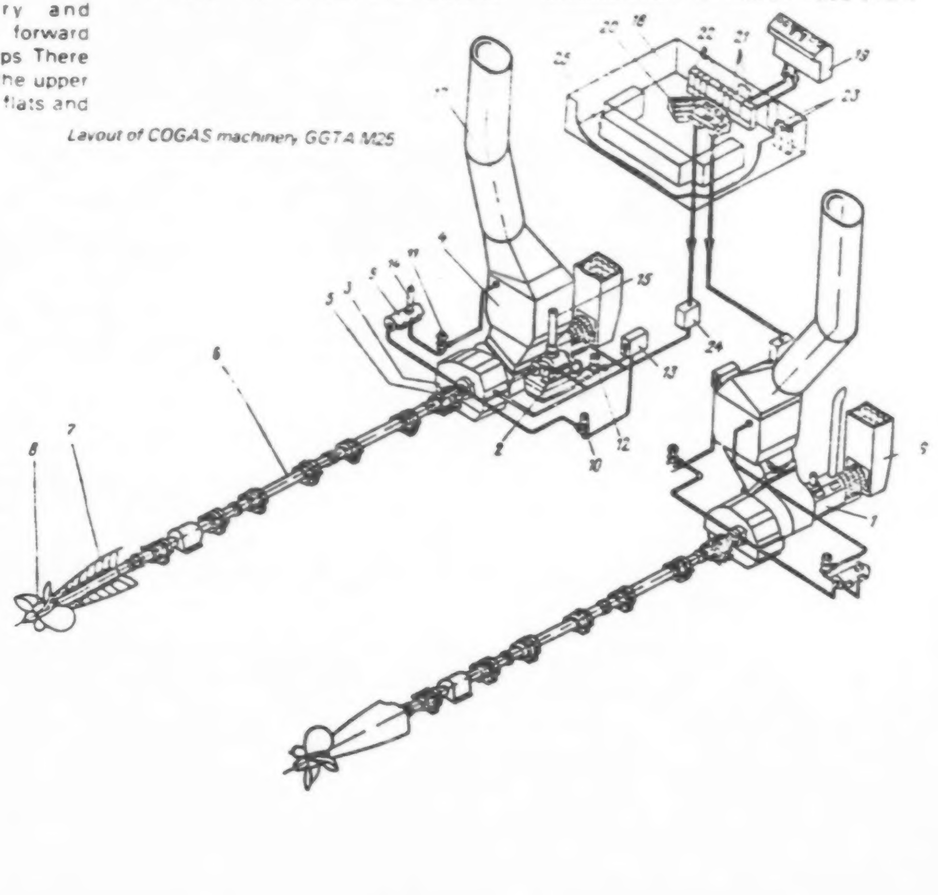
'Kapitan Smirnov' entering the lock at Tilbury *(Laurence Dun*

the ship's centreline and the steelwork weighs 150tons This ramp can accommodate differences in level between quay and the threshold to the maindeck of — 5.5 to 2.2m Thereafter all internal communication is by ramps from the main deck down to an intermediate deck and thence to the tank top by successive ramps on the port side and up by the centreline ramp between the twin engine casings to the upper tween deck and thence by a port-side ramp to the upper (weather) deck There are two large flush hatches in the upper deck forward and a portable deck for export cars Since the vessel has twin screws, the machinery and superstructure are further forward than in other large Ro-Ro ships There is a centreline passage on the upper deck leading to stowage for flats and containers aft of the superstructure

The ship's outfit includes a number of Valmet front-lift trucks, as provided in the two large Valmet-built Ro-Ro motorships of the *Magnitogorsk* class. Six Polish Towimor 12.5ton self-tensioned mooring winches are provided

The *Kapitan Smirnov* has twin-screw COGAS (combined gas and steam) turbine machinery, in which some of the heat in the high-temperature exhaust gases of the turbine are recovered and used to generate steam which is applied to complementary propulsion steam turbines. We believe this COGAS plant (often discussed in theoretical studies of the recent past) is unique i naval or merchant ship practice should be clearly distinguished fro the COSAG configuration in which the steam turbine is the base-loa machinery and which can b supplemented for higher speeds by booster gas turbine, without any thermal coupling

We have been endeavouring t obtain details of the machinery fro Sudoimport for some three years without much success, but a series of descriptive articles has appeared i our Soviet contemporary *Sudostroenie*, for March 1980 (from



Layout of COGAS machinery GGTA M25

1 *Gas turbine D158*
2 *Steam turbine PUT-2*
3 *Reduction and combining gears RO25*
4 *Gas turbine exhaust gas boiler KUP3100*
5. *Thrust block*
6. *Intermediate shaft*
7. *Stern tube with Simplex gland*
8. *Fixed-pitch propeller*
9. *Steam separator*
10. *Feed pump*
11. *Waste heat boiler circulating pump*
12. *Extraction pump*
13. *Steam chest*
14. *Steam line from separator to superheater*
15. *Steam inlet to turbine*
16. *Gas turbine air intake from plenum chamber*
17. *Gas turbine exhaust to funnel*
18. *Machinery control room*
19. *Bridge control console*
20. *Propulsion control desk*
21. *Propulsion control cubicles with synoptic display 'Shipka'*
22. *Main electrical switchboard 'Izhora'*
24. *Propulsion governor*
25. *Electrical distribution switchboard*

The principal particulars of the *Kapitan Smirnov* and the similar *Kapitan Mezhentzsev* are

Class — USSR Register	KM*L3A2
Length overall	227 3m
Length bp	204 0m
Breadth	30 0m
Depth to main deck	11 3m
Depth to upper deck	21 0m
Draught, summer	9 9m
Corresponding deadweight	20 075t
Enclosed cargo space	54 500m^3
Gross measurement	14 350t
Net measurement	7480t
Containers	1231TEU
Maximum power	36 780kW
Corresponding speed	25knots
Range	25 000nm
Ships company	39
Cadets	8
Passengers	8

which the accompanying drawings are reproduced) Particulars of the operating conditions and performance are, however, sparse

Each four-bladed 5700mm dia fixed-pitch propeller is driven by a set of machinery designated GGTA M25 with a maximum output of 18 390kW (25 755shp) at 130rev min which gives a speed of 25 knots at 9 0m mean draught At this output the specific fuel consumption is stated to be 238g kWh (0 386lb shp-h) The fuel is 280secs Marine Diesel

The gas turbines are almost certainly simple-cycle, of aero derivation, and possibly of a type already used in some classes of Soviet naval vessel They are said to contribute 14 120kW at 6000rev min, exhausting 91 8kg sec of gas at 390°C to the waste heat boiler, which produces 24 000kg h of steam at 15bar and 310°C for the steam turbine which develops 4270kW. The stack temperature is then 179°C.

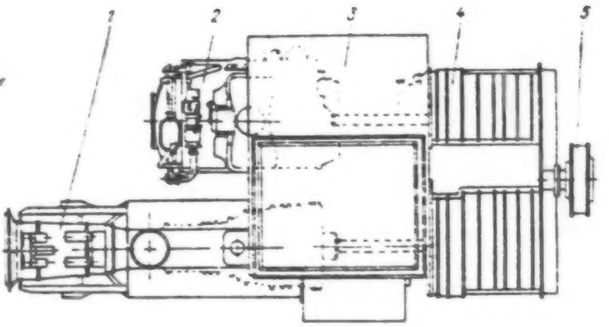
At the normal sea service output of 17 280kW (23 510shp) at 128rev min per shaft, the gas turbine develops the same 14 120kW but the steam turbine only 3160kW because part of the steam flow is diverted to power the turbo-alternators and for ship's service. The maximum astern power is 5740kW (7810bhp) per shaft The waste heat boilers cannot be *dry-fired* but the system is, to a degree, self regulating: lower gas turbine power — lower exhaust temperature and gas flow — less steam.

The transmission is somewhat complex, involving a modified form of locked-train for the gas turbine with twin idler wheels between each of the two secondary pinions and the main gearwheel (a 46-to-one reduction ratio is involved here) The steam turbine line is also double reduction, but single train, again with twin idlers in the last stage and a friction clutch between the primary wheel and the secondary shaft (presumably the steam turbine is disengaged while manoeuvring and until the waste heat boilers are generating sufficient steam)

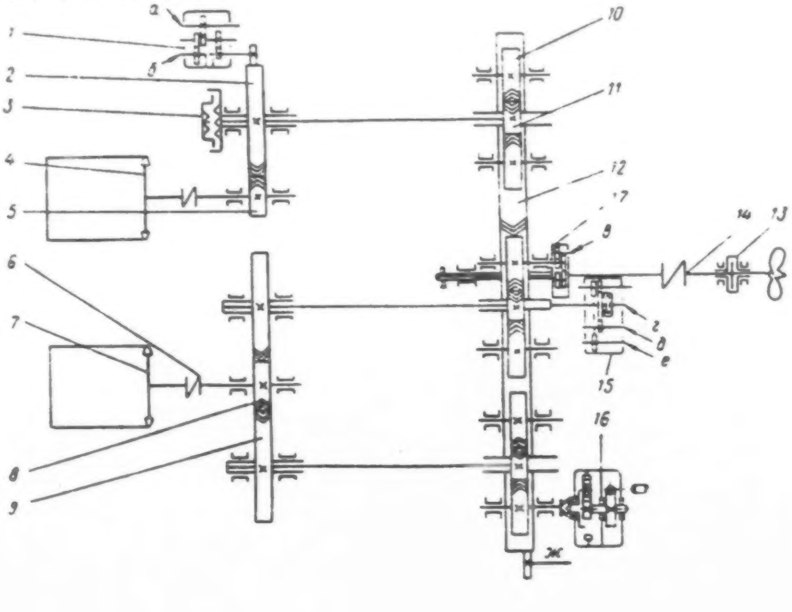
The references are quite firm about fixed-pitch propellers but it is not at all clear how the shaft reversal is effected unless the gas turbine has means for reverse rotation, quite an achievement if this is so. The astern power of 5740kW per shaft is 31 per cent of the full shaft power or 40 per cent of the gas turbine power The entire system is fully automated for control directly from the wheelhouse and there is an impressive control room containing a U-shaped console facing a comprehensive multi-panel instrument board The gas and steam turbine controls are designated BRIZ-1609 and the automation system bears the code name OMEGA

The supporting auxiliary machinery comprises three 1000kWe 750rev min diesel-alternators carried on a flat aft of the main machinery and two 1000kWe 1500rev min steam turbo-alternators which take their supply from the main waste heat boilers There are also 250kWe harbour service and emergency generator set and a 6 3t h 7 5bar oil-fired boiler The flat is sound insulated around the sides and deckhead, as the control room is above

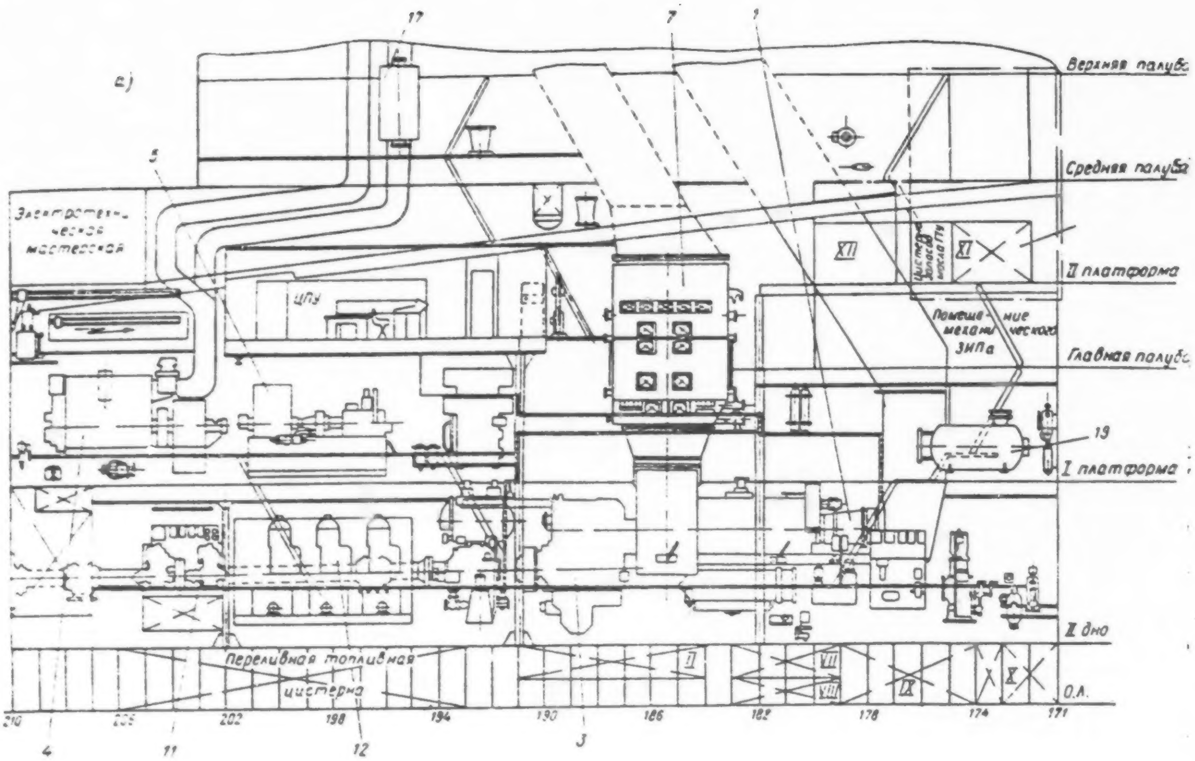
Below are the usual pumps, coolers and filters associated with turbine plant and, between the shafts the fuel treatment plant, including a Sharples module of the type described in our July 1979 issue There are two 250m^3 h ballast pumps and a 400m^3 h heel-correcting pump



Block outline of steam and gas turbine machinery for one shaft.
1 Gas turbine
2 Steam turbine
3 Waste heat boiler
4 Gearing
5 Coupling



Kinematic layout of the combining reduction gearing for one shaft Although there is a fixed-pitch propeller and only the steam turbine can be de-clutched it is not clear how astern power is achieved


а)
17
7
1
5
Верхняя палуба
Средняя палуба
Электротехническая мастерская
ЦПУ
XII
Цистерна запаса ГТУ
XI
II платформа
Помещение механического ЗИПа
Главная палуба
19
I платформа
II дно
Переливная топливная цистерна
II
VII
VIII
IX
X
О.Л.
210
206
202
198
194
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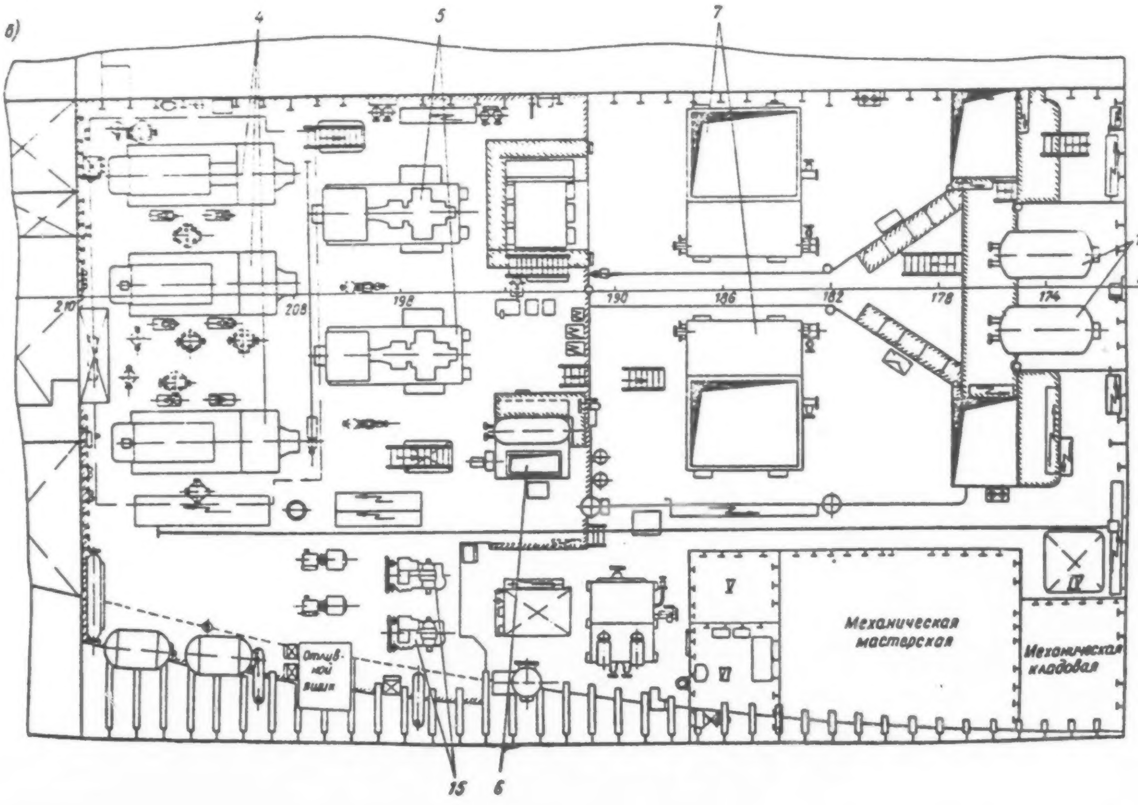
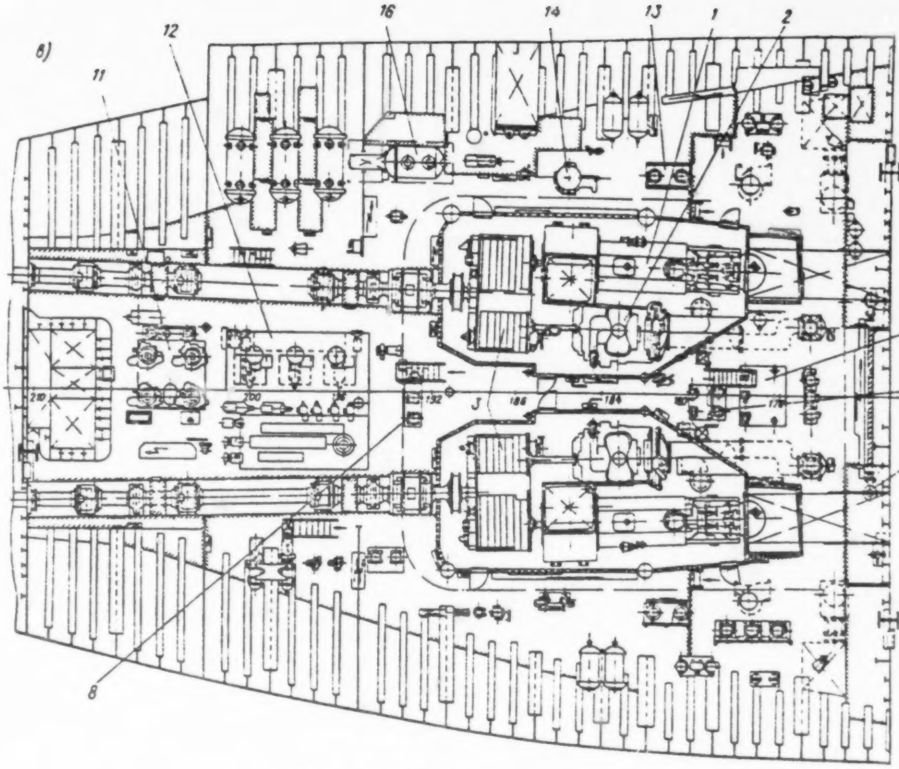
186
182
178
174
171
4
11
12
3


в)
ДП
V
VI
IV
Механическая мастерская
Механическая кладовая
Отливной ящик

Arrangement of machinery spaces. Note that the gas and steam turbines are enclosed in sound-insulating cubicles:

1. Gas turbine
2. Steam turbine
3. Gearing
4. Three 1000kWe diesel generating sets, each with six-cylinder 30/38 engine
5. Two 1000kWe turbogenerators
6. Oil-fired auxiliary boiler
7. Gas turbine waste-heat boilers
8. Lubricating oil module
9. Gas turbine fuel service pump
10. Diesel oil service pump
11. Diesel oil separators
12. Gas turbine fuel treatment module
14. Fresh water pump
15. Starting air compressors
16. Oily water separator
17. Diesel engine silencers
18. Ballast pumps
19. Steam separators


в)
16
14
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1
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11
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18
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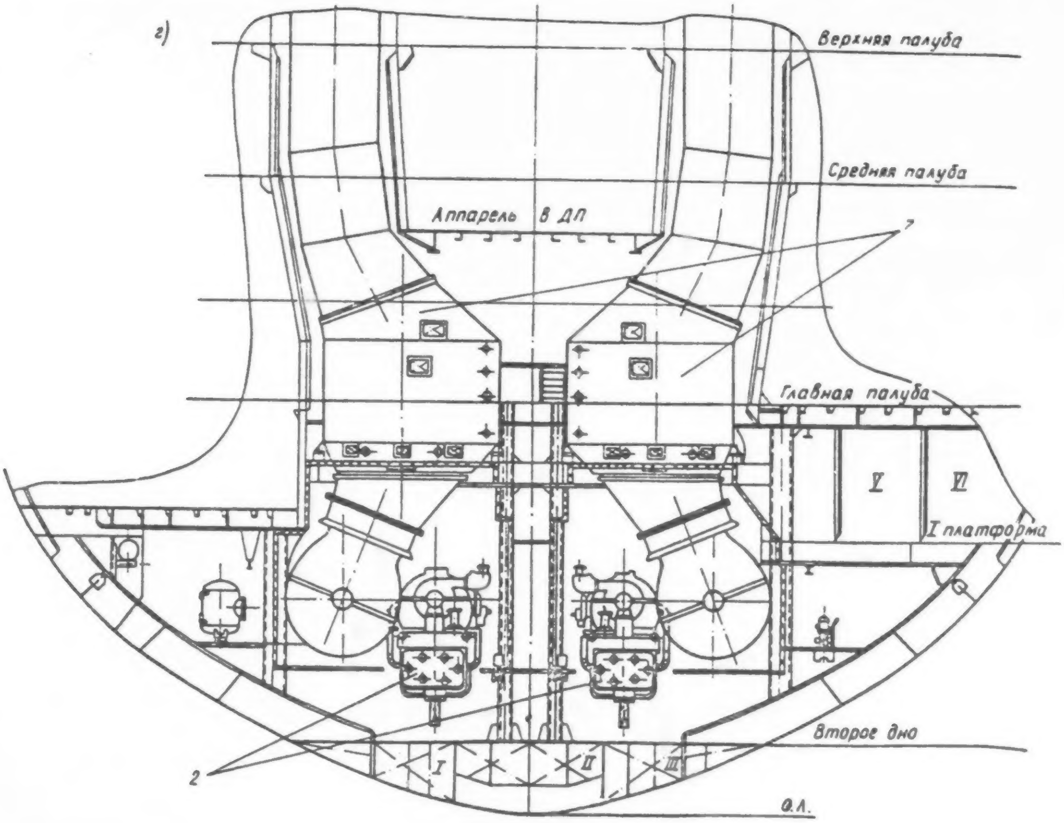
210
200
192
188
186
184

187 шп (см в нос)


г)
Верхняя палуба
Средняя палуба
Аппарель в ДП
7
Главная палуба
V
VI
I платформа
Второе дно
2
I
II
III
О.Л.

QAS

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7M'

JW

COGAS - MARINE POWER PLANT FOR ENERGY SAVINGS

By

Robert P. Giblon and I. Hilary Rolih

of

George G. Sharp, Inc.

For Presentation Before The

New York Metropolitan Section

The Society of Naval Architects and Marine Engineers

February 13, 1979

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INTRODUCTION

At the present time, the United States cannot meet its energy needs from domestic crude oil supplies and thus it is necessary to import large quantities of oil which is one of the major factors for the significant trade imbalance. Therefore, the Federal Energy Administration (FEA) has expressed the need for the U.S. merchant marine to reduce liquid fuel oil consumption, and this, toghether with the uncertainty over the future price and availability of bunker fuel and energy resources for the maritime industry as a whole, requires that the marine industry institute strong energy conservation measures in order to sustain merchant marine services, particularly in time of a national emergency.

The recent large increase in the price of marine bunker fuel oil and the potential that the price of fuel oil will continue to escalate due to the price-setting policies of the OPEC countries make it necessary for shipowners to examine and adopt ship propulsion power plant types and systems that will reduce the need for and expense of ship bunkers. It is thus the purpose of this paper to discuss one approach that may be adopted by shipowners today using known and proven technology whereby fuel energy savings of 30% to 35% over the present steam or gas turbine power plants may be obtained, as well as significant but somewhat lesser savings over diesel and present-day steam reheat power plants.

The system discussed is the combined gas-steam cycle that is widely used for shoreside electric power plants and is sometimes referred to as a STAG or COGAS cycle. This system has been chosen for study since it offers a power plant system where the all purpose fuel rate is lower than other known marine propulsion systems. The basic plant considered consists of an aircraft derivative gas turbine driving a reduction gear in tandem with a steam turbine which receives its steam from a heat recovery boiler that is located in the gas turbine exhaust gas ducting. A large portion of the waste heat from the gas turbine is thus converted into usable energy.

Although only the high efficiency aircraft derivative gas turbine has been considered , similar systems, although somewhat less efficient, can be considered using industrial type heavy duty gas turbines.

This paper has been partially based upon information developed under contract (No.7-38009) to the U.S. Department of Commerce, Maritime Administration, which was recently completed, where George G. Sharp, Inc. (Sharp) was the prime subcontractor to this government organization. This project was a joint effort and included the following subcontractors to Sharp:

Curtiss-Wright Corporation
General Electric Company, Aircraft Engine Group
Foster Wheeler Boiler Corporation
Sun Shipbuilding & Dry Dock Company

The participation of the above organizations in this project was important because of their contribution in the following areas of interest:

- Curtiss-Wright Corporation: Power plant system integration and marketing; manufacture and supply of reduction gears
- General Electric Company - Aircraft Engine Group: Marine propulsion aircraft derivative gas turbines and auxiliary equipment, such as engine starting systems and controls
- Foster Wheeler Boiler Corporation: Shoreside COGAS installations, manufacture of heat recovery boilers and steam condensers
- Sun Shipbuilding & Dry Dock Company: Overview of the propulsion power plant system from a shipbuilder's point of view due to their experience in the installation of aircraft derivative type gas turbine machinery in a large ocean-going, commercial type ship, the CALLAGHAN. Sun Ship also estimated the differentials in cost for installing the COGAS plan in a ship as compared with a conventional steam plant.

The results of the study indicate:

- Overall fuel rates of about 0.31 lbs/shp-hr can be obtained when burning marine gas oil or clean marine diesel oil when the plant is delivering about 30,000 shp.

- Capital costs are lower for the COGAS plant as applied to ships where auxiliary steam loads are low such as in containerships, but costs are higher where the auxiliary steam loads are high such as on tankers.

- Lower shipboard maintenance requirements and a simpler power system permit smaller engine room crews for the COGAS plant resulting in savings in crew and subsistence costs.

- Use of one gas turbine model, the General Electric LM-2500, coupled with the use of standardized machinery components, allows one COGAS plant design to be used for power ranges of 20,000 to 36,700 shp.

- Due to the higher price of distillate fuels, the COGAS plant offers only nominal fuel cost savings to the commercial steamship operator burning heavy lower price fuel. However, this situation could change if testing now underway proves that treated blended fuel (IF 60) can be burned.

- Installation of a COGAS plant in U.S. naval auxiliary ships can offer significant savings in yearly fuel cost, crew size and cost, crew training and logistics since the same type of gas turbine now fitted in new combatant ships can be utilized. The cost differential in fuels burned does not exist for naval ships since the same distillate fuel is burned in steam, diesel and gas turbine ships.

WHAT IS A COGAS SYSTEM?

The COGAS system is a Combination Of a Gas turbine And a Steam turbine to produce propulsion power. The gas turbine is usually of the open cycle type although a regenerative gas turbine may be utilized. The exhaust gas from the turbine is directed to an unfired heat recovery heat exchanger (boiler) and the steam produced by this unit is passed through a steam turbine. The steam turbine exhaust is rejected to a condenser where the resulting condensate is collected and pumped back into the heat exchanger. Figure 1 illustrates the COGAS system in its simplest form. The gas and steam turbines are geared together, driving the propeller via a double reduction gear. Since the gas turbine usually rotates in only one direction, a means for reversing the propeller thrust must be provided such as reversing gears or a controllable, reversible pitch propeller.

Auxiliary electric power may be provided by separate independent generators or by a steam turbogenerator supplied with steam from the heat recovery boiler or, alternatively, the generator can be driven from the main propulsion gear train. In any event, separate independent generators are required for in-port and maneuvering conditions. The heat source for ship's heating, operation of the evaporator and similar type ancillary equipment can be from an electrical power source or can be from the heat recovery boiler, or a combination of both. Where the heat source is taken from a heat recovery boiler, an independent auxiliary oil-fired boiler is required for use in port and for maneuvering operations.

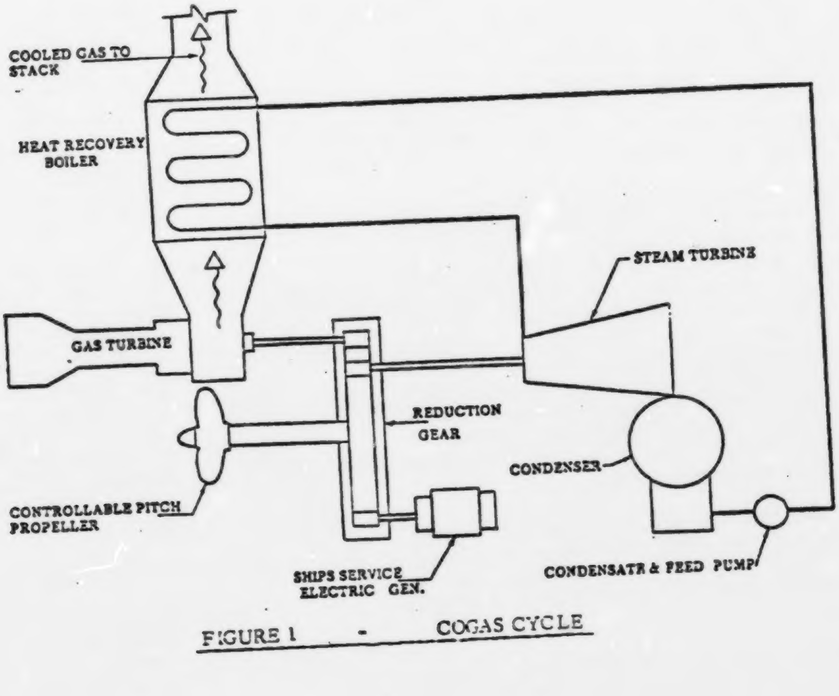


FIGURE 1 - COGAS CYCLE

COGAS CYCLE AND PERFORMANCE

Prior to undertaking the contract with the government for the COGAS study Sharp selected the General Electric LM-2500 open cycle, marinized, gas turbine since this engine is very energy-efficient and is widely used in marine applications, particularly by our own U.S. Navy. Preliminary studies that were conducted in late 1975 indicated that, not only was the COGAS cycle using the LM-2500 very energy-efficient, but it could also offer significant savings to the shipowner in yearly fuel cost when burning higher-priced distillate fuels when compared to the conventional present-day steam plant burning lower cost Bunker C (No.6) residual fuel oil. Unfortunately, as the study progressed, present government policies that encourage the production of residual fuel restricted price increases for this fuel even though crude oil prices were rising due to the greater foreign crude import, while distillate prices were rising rapidly. The net result was that the initial advantage anticipated for the COGAS plant as it relates to yearly fuel cost was considerably reduced.

A cross-sectional view of the General Electric LM-2500 gas turbine is shown in Figure 2. This machine is normally mounted in a sound attenuating module as shown in Figure 3 with the air inlet (inlet plenum) on one end and the turbine exhaust on the other. In general the gas turbine engine is completely self-contained except for some separate support equipment required for starting, monitoring and control.

The performance points selected, such as power output, ambients, pressure losses, etc., for the gas turbine on which it was decided to make all calculation and comparisons are as noted in Table 1. This rating was selected since it would result in a total COGAS power plant output of about 30,000 shp. Further the gas turbine rating at 22,620 bhp was considered conservative and would result in relatively long life for the engine hot end section and thus result in a minimum of maintenance. The maximum continuous power rating for the gas turbine engine alone is about 31,000 bhp.

TABLE 1

COGAS POWER PLANT PERFORMANCE
BASED ON

Gas Turbine Type	GE LM-2500
Gas Turbine Maximum Continuous Rating, SHP	22,168
Reduction Gear Loss, %	2
Turbine, BHP	22,620
Engine, RPM	3,600
Inlet Air Temp, F	75
Relative Humidity, %	85
Inlet Loss, Water Inches	3.0
Outlet Loss, Water Inches	12.8
Gas Turbine Inlet Temp, F	1,411
Gas Turbine Exhaust Temp, F	966
Pounds per hour Exhaust Gas	468,720
Fuel Consumption, lbs/hr.	9,315
Fuel Rate, lbs/shp-hr.	0.4118
Type Fuel	#2 (Gas Oil)
Fuel, Lower Heating Value, btu/lb.	18,120

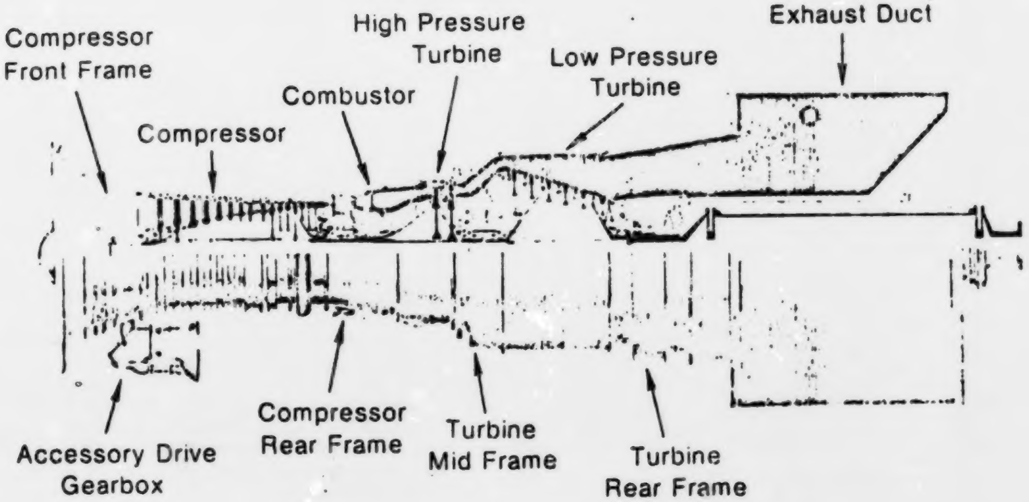


FIGURE 2 - MARINE GAS TURBINE ENGINE (LM2500)

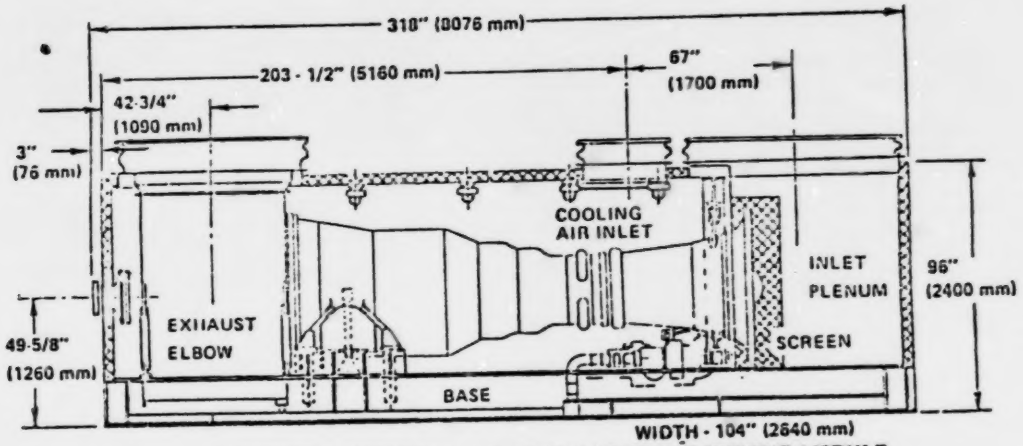


FIGURE 3 - GENERAL ELECTRIC LM2500 GAS TURBINE MODULE

To determine the best COGAS heat cycle arrangement, optimization studies were conducted during the preliminary design phase of the project and these included the following topics:

- Establishment of preliminary COGAS heat cycles based on gas turbine performance information from the manufacturer. The heat cycles studies covered a range of steam pressures and superheat temperatures, condenser vacuums and heat recovery boiler terminal temperature differences. Steam cycle performances for both single pressure and multi-pressure heat recovery boilers were analyzed by using a computer program developed to carry out the heat balance calculations.
- Determination of performance for single and multi-pressure boilers based on information from the heat recovery manufacturer, Foster Wheeler Boiler Corp. (Foster Wheeler). For the various boilers considered, information included basic design parameters as well as cost and weight differentials. The boilers covered a range of three different steam pressures and superheater outlet temperatures for various pinch points (differences in temperature between exhaust gases at the generating bank outlet(s) and saturated temperature at drum pressure(s)) and economizer approaches (degrees below saturation of economizer water outlet).
- Determination of the steam turbine manufacturer's performance, weight and cost differential data for the established COGAS heat cycles, including steam induction into the lower pressure stages. Cost and performance information was obtained from General Electric (Lynn) and the Terry Corporation.
- Determination of the condenser size, weight and cost. Weight and cost information was furnished by Foster Wheeler.
- Determination of the basic limits for the fuels which can be economically burned in the aircraft derivative gas turbine. Fuel costs and availability were investigated for ports in the U.S.A. and abroad.

The COGAS cycles to be studied were first divided into two groups; those using single pressure boilers, and those with multi-pressure boilers. Descriptive cycle diagrams of these two schemes are shown on Figures 4 and 5.

It will be noted that both cycles selected include ship's service electric generators attached to and driven by the main reduction gear. This arrangement was selected since the attached generator is more efficient than a separate turbo-generator driven by steam from the heat recovery boiler or separate diesel generators. The cycle also provides steam for the evaporators to produce water for the ship's crew needs and make-up water for the main propulsion plant.

The merits of the various alternative COGAS cycles were carried out by comparing capital costs and fuel economy with those for a base cycle. The base cycle used for the purpose of conducting the screening was a single pressure boiler cycle with throttle conditions of 450 psig/850F, condenser pressure of 2" Hg abs, and a 30F pinch point. For further details, see Appendix A "Economic Evaluation of COGAS Cycles". The COGAS cycle that was finally selected as a result of the system optimization studies, however, is a single pressure boiler cycle with throttle conditions of 450 psig/885F, condenser pressure of 2" Hg abs and an 18F pinch point. For details of performance, see Appendix B.

The steam turbine produces about 30% of the propulsion power as indicated in Table 2. Since the gas turbine is non-reversible, the steam turbine is similarly designed, that is, it is not fitted with an astern element.

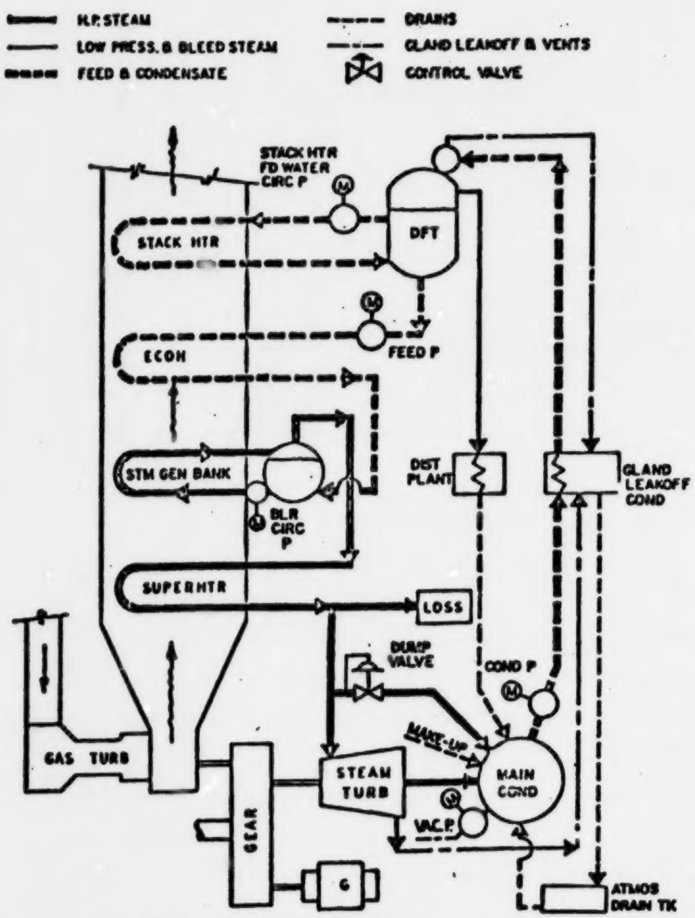

H.P. STEAM
LOW PRESS. & BLEED STEAM
FEED & CONDENSATE
DRAINS
GLAND LEAKOFF & VENTS
CONTROL VALVE
STACK HTR FD WATER CIRC P
M
DFT
STACK HTR
ECON
FEED P
STM GEN BANK
BLR CIRC P
DIST PLANT
GLAND LEAKOFF COND
SUPERHTR
LOSS
DUMP VALVE
COND P
MAKE-UP
GAS TURB
STEAM TURB
MAIN COND
VAC.P.
GEAR
G
ATMOS DRAIN TK

FIGURE 4 - COGAS HEAT CYCLE
WITH SINGLE PRESSURE HEAT RECOVERY BOILER

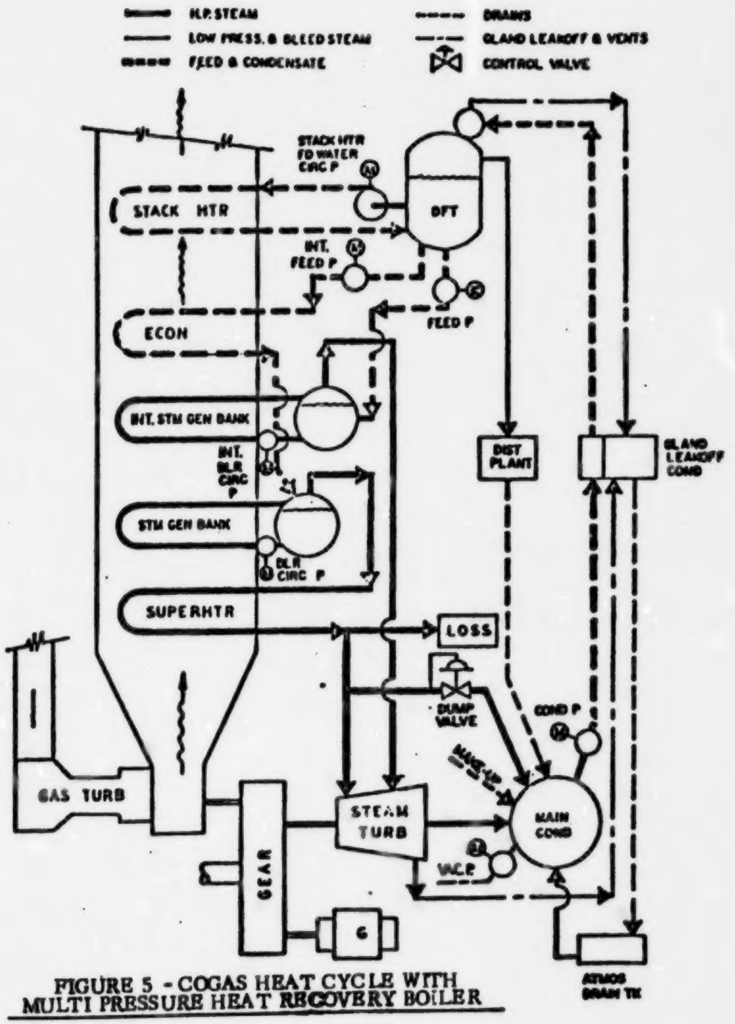

H.P. STEAM
LOW PRESS. & BLEED STEAM
FEED & CONDENSATE
DRAINS
GLAND LEAKOFF & VENTS
CONTROL VALVE
STACK HTR FD WATER CIRC P
M
DFT
STACK HTR
INT. FEED P
FEED P
ECON
INT. STM GEN BANK
INT. BLR CIRC P
DIST PLANT
GLAND LEAKOFF COND
STM GEN BANK
BLR CIRC P
SUPERHTR
LOSS
DUMP VALVE
COND P
MAKE-UP
GAS TURB
STEAM TURB
MAIN COND
VAC.P.
GEAR
G
ATMOS DRAIN TK

FIGURE 5 - COGAS HEAT CYCLE WITH
MULTI PRESSURE HEAT RECOVERY BOILER

The reduction gear arrangement consists of two separate components: viz, a conventional combining/reduction gear to which the two turbines are connected, acting as a first reduction gear driving a planetary second reduction gear through an intermediate shaft. The planetary gear contains a thrust bearing and drives the propeller through a low-speed shaft. Figures 6 and 7 illustrate the type of planetary gear considered.

Although a planetary type gear was selected in the studies for the second reduction gear, the conventional type first and second reduction gear, all contained in one gear box, could have been selected.

TABLE 2

STEAM TURBINE PERFORMANCE

Type Turbine	Single Casing
Turbine Power, BHP	9,123
Power to Generator, Shaft, BHP	1,185
Power Propeller Shaft, SHP	7,777
Turbine, RPM	5,500
Inlet Pressure, PSIG	450
Inlet Temperature, F	885
Condenser Pressure, ins. hg. abs	2.0
Turbine Steam Rate, lbs/shp-hr. 7.167	6.27 – .96% r.
Steam Flow, lbs-hr.	56,196

The combining gear box is fitted with a take-off shaft which is used to drive a ship's service generator which is normally in use under steady state conditions. Standby, maneuvering and start-up conditions utilize electric power from one of two diesel generator sets.

Further information on the reduction gears is given in Table 3.

To obtain maximum heat recovery and efficiency, the steam plant is "slaved" to the gas turbine, allowing the steam pressure to vary in response to changes in the gas turbine exhaust conditions. Figure

TABLE 3

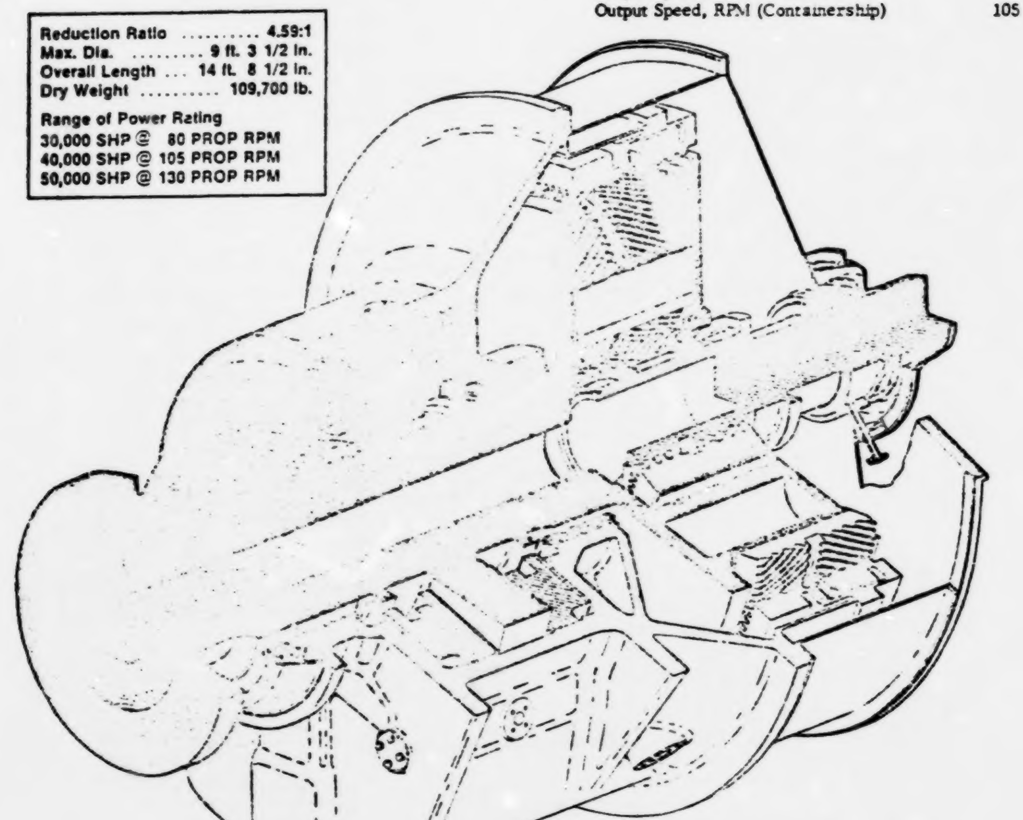
REDUCTION GEARS
NOMINAL POWER OUTPUT 30,000 SHP

First Reduction Module

Type	Combining, Single Reduction
Gas Turbine Speed, RPM	3,600
Steam Turbine Speed, RPM	5,500
Generator Speed, RPM	1,800
Output Speed, RPM (Tanker)	370
Output Speed, RPM (Containership)	480

Second Reduction Module

Type	Planetary
Output Speed, RPM (Tanker)	80
Output Speed, RPM (Containership)	105



Reduction Ratio **4.59:1**
Max. Dia. .... **9 ft. 3 1/2 in.**
Overall Length ... **14 ft. 8 1/2 in.**
Dry Weight .... **109,700 lb.**

Range of Power Rating
30,000 SHP @ 80 PROP RPM
40,000 SHP @ 105 PROP RPM
50,000 SHP @ 130 PROP RPM

FIGURE 6 - CURTISS WRIGHT MODULE "A" MARINE PLANETARY TRANSMISSION

PRELIMINARY INSTALLATION SKETCH
PLANETARY TRANSMISSION ASSEMBLY

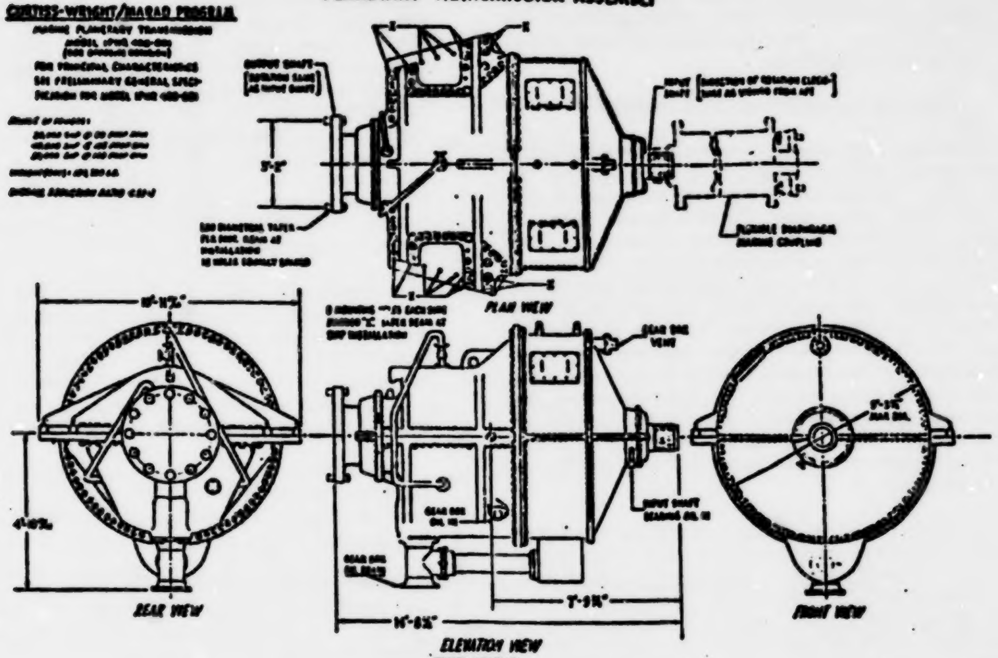
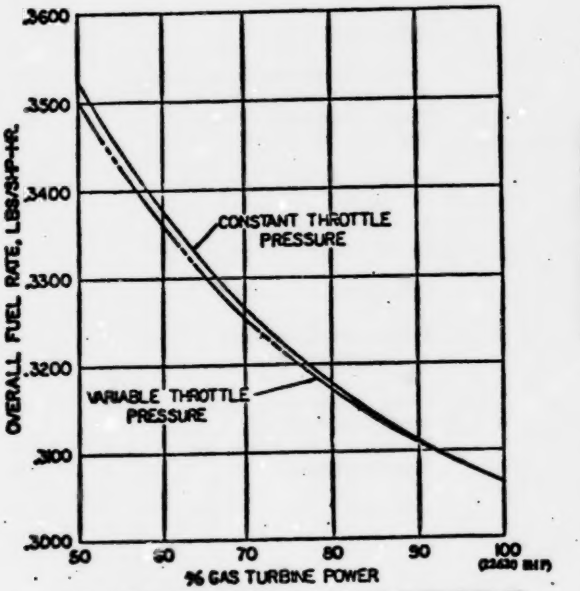
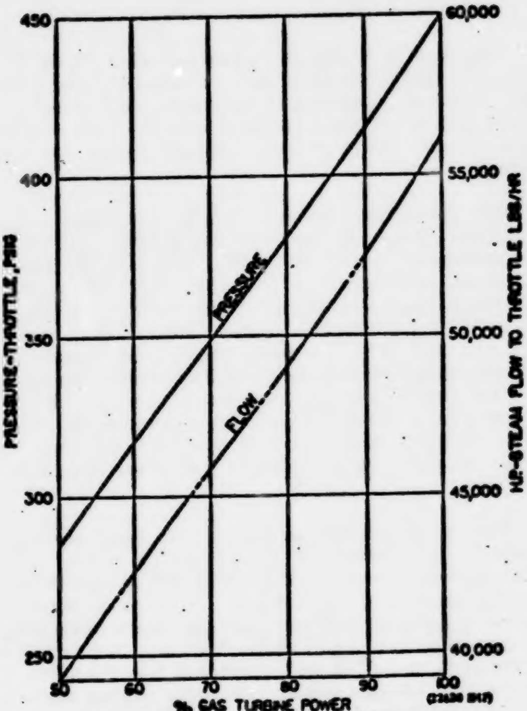

CURTISS-WRIGHT/MARAD PROGRAM
MARINE PLANETARY TRANSMISSION
5'-2"
OUTPUT SHAFT
INPUT SHAFT
PLAN VIEW
GEAR BOX VENT
REAR VIEW
FRONT VIEW
ELEVATION VIEW
7'-9¾"
4'-11⅞"
16'-8¾"

FIGURE 7



FULL GAS TURBINE SPEED TO 50% GAS TURBINE POWER

FIGURE 8 - FUEL RATE CURVES
CONSTANT VS VARIABLE THROTTLE PRESSURE



FULL GAS TURBINE SPEED-50% GAS TURBINE POWER
285 PSIG-DUMP OPENS-70% STEAM FLOW

FIGURE 9 - THROTTLE PRESSURE AND
FLOW VS G.T.POWER

8 shows the advantage of using the variable steam turbine throttle pressure versus constant pressure, while Figure 9 shows the throttle steam pressure and flow as a function of the percentage of total power produced by the gas turbine.

HEAT RECOVERY BOILER DESIGN

The unfired heat recovery boiler is located in the exhaust stream from the gas turbine and generates the steam for the steam propulsion turbine at about 460 psig and 890F when the COGAS power plant is delivering about 30,000 shp. Steam pressure and flow is a function of the gas turbine exhaust conditions, which vary primarily due to changing power conditions, and, therefore, the steam plant response will vary according to the heat recovery from the gas turbine exhaust. The heat recovery boiler also delivers steam to the deaerating feed heater and the evaporator via a separate loop at about 10 psig. Further details on the selected heat recovery boiler as designed by the Foster Wheeler Boiler Corp. are noted in Table 4. The general configuration of the boiler is shown in Figures 10a, 10b and 10c.

From Table 4 and Figure 10a-c it will be noted that the boiler is similar to that used for the average boiler economizer on steamships burning heavy residual fuel. Since clean distillate fuel is to be burned in the COGAS propulsion plant, long life with minimum maintenance can be expected from this rugged type of construction. In general, maintenance on the heat recovery boiler should be considerably less than the conventional economizer used in a steam plant.

Although it is not expected that cleaning of the heat exchanger surfaces will be required very often due to the wide spacing of the tubes and fins and the burning of relatively clean distillate fuel, nevertheless soot blowers will be provided between the various tube banks.

The generating portion of the heat recovery boiler supplying steam to the main propulsion turbine is force-circulated by means of a pump. To assure adequate circulation a standby pump will automatically start in event of failure of the main pump. The use of forced circulation in steamship boilers is not new but very few have been fitted. However, forced-circulation boilers are in common use shoreside and are almost universally used on COGAS (and STAG) power plants. Since the pump head pressure and steam drum pressure are relatively low, no problems are expected from the inclusion of the forced-circulation feature.

The stack heater portion of the heat recovery boiler is also fitted with a circulating pump. This heater serves the deaerating feed heater which in turn supplies steam to the distiller. The pump supplies water to the heat exchange surface at the deaerating feed tank temperature (240F). This inlet temperature is the same as used in most pre-World War II and World War II type steam vessels. Since the sulphur content of the fuel burned will be low, it is expected that there will be little or no corrosion in the cold end of the unit but in any event it will be considerably less than that experienced in economizers on World War II type vessels.

The gas temperature leaving the stack heater is expected to reach about 310F at 30,000 shp.

The COGAS plant can be operated with the gas turbine secured. For commercial tanker installations, the steam turbine can be supplied with steam from the auxiliary boiler that would be fitted to drive cargo pumps and perform tank heating services. For containerships or similar type vessels where the auxiliary boiler may be small, the heat recovery boiler will be heated by an emergency combustor that will supply hot gas at about the same temperature as would come from the gas turbine exhaust (950F). Air to the combustor would be supplied by a motor driven forced draft fan. Figure 11 is a view of the emergency combustor.

A significant feature of the COGAS propulsion plant is its ability to operate on the gas turbine alone with all the gas turbine exhaust passing through the heat recovery boiler. The heat recovery boiler may be operated dry and, unlike a conventional steam boiler, it can withstand complete loss of feed water without causing any damage to any of the tube banks.

The flexibility inherent in operating the COGAS plant on either the gas or steam turbine alone provides higher reliability and permits emergency maintenance at sea on the unit out of service.

For additional information relating to operation of the plant with either the steam or gas turbine secured, the reader is referred to the section on Control Systems which follows.

TABLE 4

SELECTED HEAT RECOVERY BOILER

Performance	
Steam, Superheater Outlet, lbs/hr.	56,522
Pressure, Superheater Outlet, PSIG	462
Temp., Superheater Outlet, F	890
Gas Temp. to Unit, F	966
Recirculated Flow Stack Heater, lbs/hr.	8,776
Steam Pressure Heater, PSIG	10
Uptake Temperature, F	311
Draft Loss, ins. water	10
Economizer Superheater & Steam Generating Section	
Tubes, in line pitch, ins.	4-1/2
Tube, O.D., ins.	2
Fin Height & Thickness, ins.	3/4-0.105
Fins/Foot (about)	36

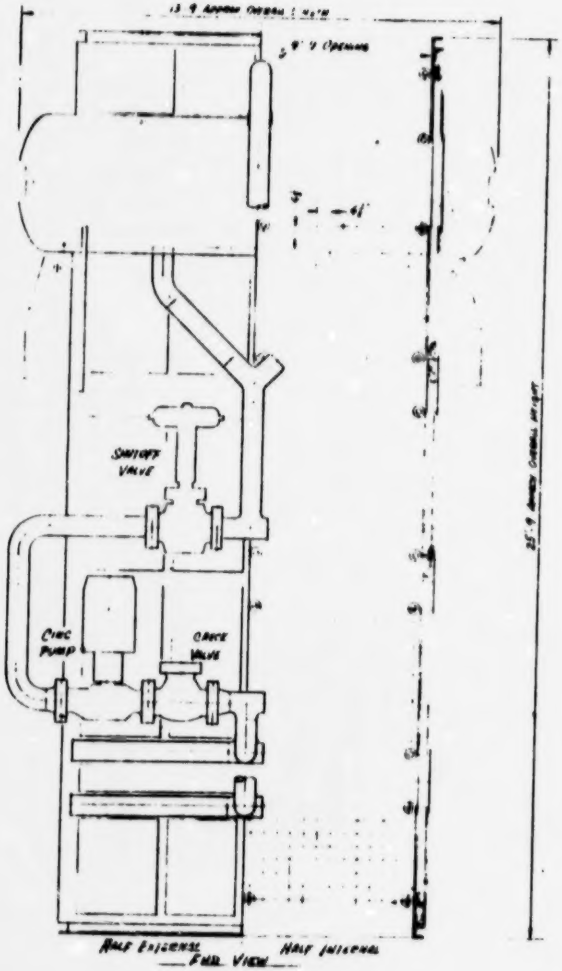

13' 9 Approx Overall Width
3' 9 Opening
Shutoff Valve
Circ Pump
Check Valve
25' 9 Approx Overall Height
Half External
End View
Half Internal

FIGURE 10a

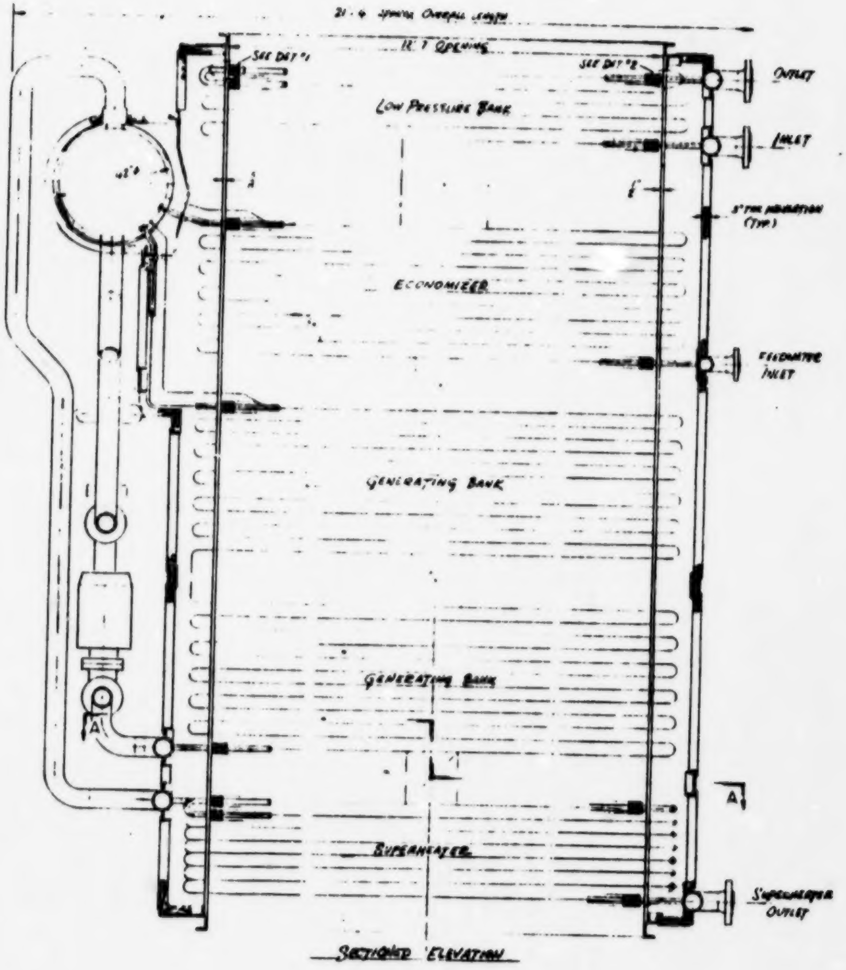

21' 4 Approx Overall Length
12' 7 Opening
See Det #1
See Det #2
Outlet
Inlet
Low Pressure Bank
3" Pipe Insulation (Typ)
Economizer
Feedwater Inlet
Generating Bank
Generating Bank
Superheater
Superheater Outlet
A
A
Sectioned Elevation

FIGURE 10b

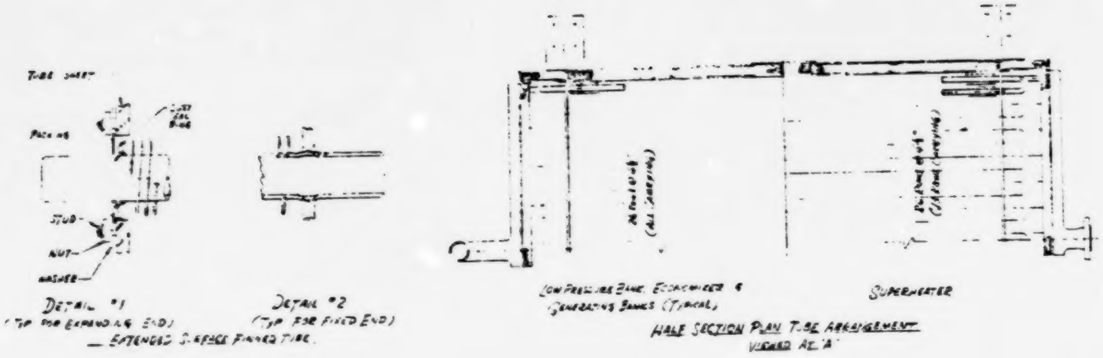

TUBE SHEET
PACKING
STUD
NUT
WASHER
DETAIL #1
(TYP FOR EXPANDING END)
DETAIL #2
(TYP FOR FIXED END)
EXTENDED SURFACE FINNED TUBE
LOW PRESSURE BANK, ECONOMIZER &
GENERATING BANKS (TYPICAL)
SUPERHEATER
HALF SECTION PLAN TUBE ARRANGEMENT
VIEWED AT 'A'

FIGURE 10c

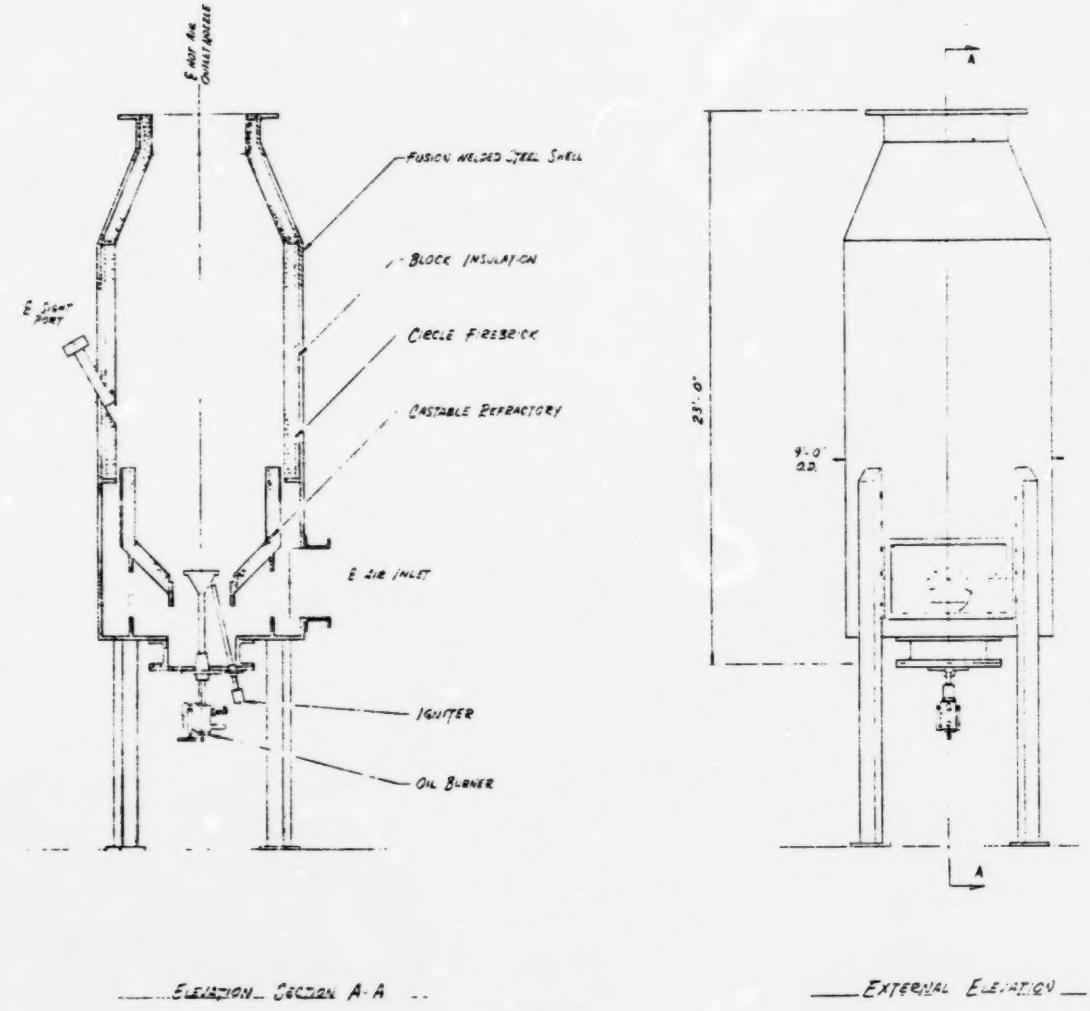

℄ HOT AIR OUTLET NOZZLE
FUSION WELDED STEEL SHELL
BLOCK INSULATION
CIRCLE FIREBRICK
CASTABLE REFRACTORY
℄ SIGHT PORT
℄ AIR INLET
IGNITER
OIL BURNER
A
23'-0"
9'-0" O.D.
A
ELEVATION SECTION A-A
EXTERNAL ELEVATION

FIGURE 11

CONTROL SYSTEMS

The Control Concepts have been conceived around the following basic principles:

- Single lever bridge control and unattended engine room operation during steady state, maneuvering and standby periods.
- High reliability and low cost by maximizing use of proven control methods and concepts, and by simplifying control loops where possible. Where adaptable, component manufacturer standard control systems will be used.
- Maximum heat recovery by "slaving" the steam plant to the gas turbine, allowing steam pressure to vary in response to changes in gas turbine exhaust gas conditions.
- The use of a controllable pitch propeller to enable reversing and maneuvering without the necessity for clutches.
- Maximum availability of a gear-driven generator, requiring limited shaft speed variation.

The control system for the COGAS plant with its controllable pitch propeller is shown in Figure 12. The steam portion of the control will normally be "slaved" to its "master" gas turbine and pitch control.

The control levers for speed and directional control will be mounted on the bridge with a secondary lever on the control console in the engine room. One or the other control position is selected by the operator.

A mode selection switch is mounted adjacent to each of the control levers. It has three positions which govern the choice of speed/power program and electric power source as follows:

- In the maneuvering mode electricity is supplied by a diesel generator, and propeller pitch and shaft speed are optimized for rapid response.
- In the seagoing mode, electricity is supplied by a gear-driven generator. The speed/power program keeps shaft speed constant and varies the propeller pitch in the 50-100% ship power ranges.
- Emergency operation is described later. In certain modes, operation may continue with single lever control, using a different speed/power program.

The function generators, one for gas turbine power and one for propeller pitch angle, will receive the linear speed/direction command from the selected control lever and generate gas turbine and propeller pitch commands. These commands will be generated by the selected speed/power program which will set the optimum combination of gas generator speed and propeller pitch angle within the limits of the operating mode.

The speed/power programs will take into account the large differences in the system time constants between the gas and the steam turbine cycles. The system will also optimize the programming of the gas turbine fuel control and the propeller pitch angle control to provide acceptable transient response and steady state operating characteristics for the COGAS plant.

The gas turbine controls are those normally furnished with the General Electric LM-2500 engine. The system receives a power command from the function generator, together with pressure, temperature and speed signals fed back from the gas turbine and to control fuel flow and gas generator speed. The LM-2500 control system adjusts fuel flow rate, and positions the compressor inlet guide vanes and stator vanes in response to operating conditions, to produce the required power turbine output power and speed.

The propeller pitch servo control unit is the standard system supplied by the propeller manufacturer. It receives the pitch command from the pitch control function generator and strokes the blade pitch actuator until the desired pitch is achieved, as indicated by the pitch position signal fed back from the blades.

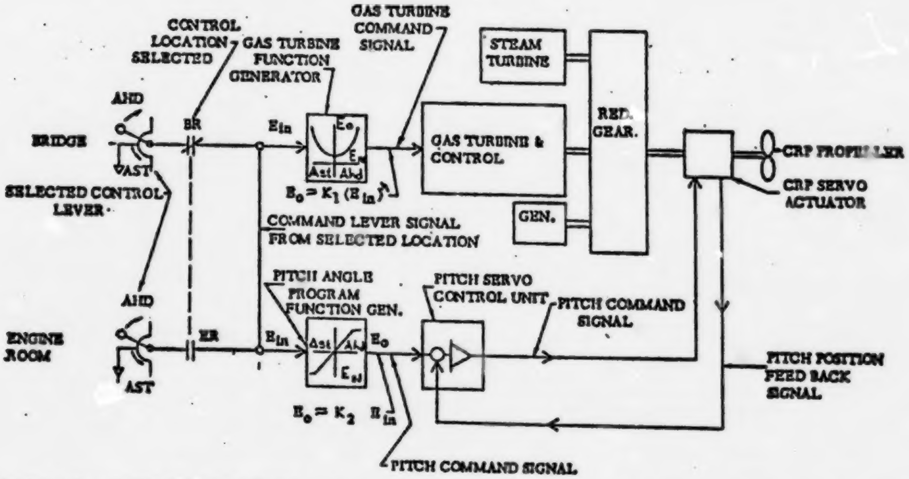

CONTROL LOCATION SELECTED
GAS TURBINE FUNCTION GENERATOR
GAS TURBINE COMMAND SIGNAL
STEAM TURBINE
AHD
BRIDGE
AST
BR
E_{in}
E_o
Ast
Ahd
$E_o = K_1 (E_{in})^2$
GAS TURBINE & CONTROL
RED. GEAR.
CRP PROPELLER
CRP SERVO ACTUATOR
SELECTED CONTROL LEVER
COMMAND LEVER SIGNAL FROM SELECTED LOCATION
GEN.
PITCH ANGLE PROGRAM FUNCTION GEN.
PITCH SERVO CONTROL UNIT
PITCH COMMAND SIGNAL
AHD
ENGINE ROOM
AST
ER
E_{in}
E_o
E_{st}
$E_o = K_2 \; E_{in}$
PITCH POSITION FEED BACK SIGNAL
PITCH COMMAND SIGNAL

FIGURE 12 - COGAS SIMPLIFIED CONTROL SYSTEM DIAGRAM - BASIC GAS TURBINE AND CRP PROGRAM

The steam plant control system is not an integrated system but consists of discrete conventional automatic controls for each of the various plant functions. These are illustrated in Figure 13 and described below:

a) Boiler pressure is not controlled directly, but is determined by the temperature and flow rate of the gas turbine exhaust gases. Upper limits of boiler pressure are established by the steam dump valve and by the boiler and superheater safety valves.

b) Superheater temperature is not controlled directly, but is determined by the temperature of the gas turbine exhaust gases and the heat transfer characteristics of the system. No temperature control is necessary for protection of superheater or turbine.

c) The turbine admission valve is wide open in the seagoing mode, allowing the turbine to consume all steam generated by the heat recovery boiler. The valve closes in the maneuvering mode automatically signalling the opening of the steam dump valve.

The admission valve acts as a safety shut off valve and speed limiting device. The valve is spring loaded and oil pressure maintained in accordance with the steam turbine manufacturer's standard practice. The valve will close on low lube oil pressure and high drum water level. Low lube oil pressure in the steam turbine system will activate an "emergency shut-down" button.

d) Excess boiler pressure is relieved directly to the main condenser through a diaphragm operated back pressure valve and orifice set to open before the boiler safety valves. At all times that the turbine admission valve is closed, a solenoid is activated to jack open the dump valve to assure a flow of steam through the superheater and main steam line. By arranging the dump line from a low point in the steam line, and with a slope to the condenser, positive drainage of the steam line, of the turbine admission valve, and of the superheater is assured at all times.

e) A ventilating steam supply to the turbine, which could be activated by the position of the turbine admission valve, is not thought to be necessary.

f) Deaerating feed tank (DFT) level control is controlled only at the extreme ends of the permissible range. A low level signals the opening of a make-up feed valve from the distilled water tank to the main condenser. A high level opens a dump valve from the DFT to the same tank.

g) DFT shell pressure is maintained constant by circulation of water through the stack heater by a motor driven pump. Excess pressure is relieved by a diaphragm operated valve to the main condenser. Low pressure signals the opening of a live steam supply valve.

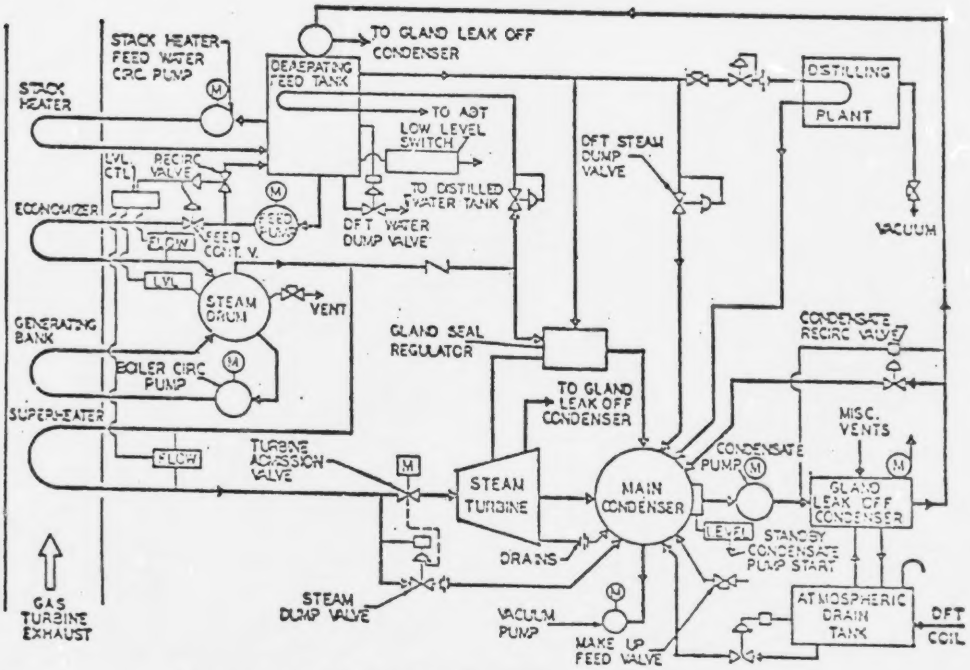

STACK HEATER FEED WATER CIRC PUMP
TO GLAND LEAK OFF CONDENSER
DEAERATING FEED TANK
STACK HEATER
M
TO ADT
LOW LEVEL SWITCH
LVL. CTL
RECIRC VALVE
TO DISTILLED WATER TANK
ECONOMIZER
FEED PUMP
FLOW
FEED CONT. V.
DFT WATER DUMP VALVE
DFT STEAM DUMP VALVE
DISTILLING PLANT
VACUUM
LVL
STEAM DRUM
VENT
GENERATING BANK
GLAND SEAL REGULATOR
BOILER CIRC PUMP
CONDENSATE RECIRC VALVE
TO GLAND LEAK OFF CONDENSER
SUPERHEATER
MISC. VENTS
FLOW
TURBINE ADMISSION VALVE
CONDENSATE PUMP
STEAM TURBINE
MAIN CONDENSER
GLAND LEAK OFF CONDENSER
LEVEL
STANDBY CONDENSATE PUMP START
DRAINS
GAS TURBINE EXHAUST
STEAM DUMP VALVE
VACUUM PUMP
MAKE UP FEED VALVE
ATMOSPHERIC DRAIN TANK
DFT COIL

FIGURE 13 - COGAS POWER PLANT STEAM CYCLE

h) Stack temperature is not controlled directly, but is prevented from falling by maintenance of DFT shell pressure, hence stack heater surface temperature.

i) Boiler drum water level is maintained in the steam drum by a conventional three element control, sensing drum level by a standpipe, steam flow by pressure drop across the superheater, and feedwater flow by the pressure drop across the economizer. The system controls the feed regulating and recirculating valves.

When the water level in the drum reaches a pre-set high level, the turbine admission valve will close causing the steam dump valve to open.

j) Feedwater flow rate to the boiler from the constant speed, centrifugal feed pump is controlled by the feedwater control valve and the feedwater recirculating valve action. Both receive an air signal from the three element control system which reduces the flow to the boiler by throttling the feedwater control valves and activates the feedwater recirculating valve such that it is full open at approximately 10 percent of the feed pump rated capacity.

k) Gland sealing steam pressure for the steam turbine is maintained within 0.5 psi of 2.0 psig by a conventional gland sealing steam regulator, dumping excess pressure to the condenser and supplying steam from the DFT.

l) The atmospheric drain tank (ADT) level is maintained above a minimum depth by a differential level, air pilot-operated valve to prevent air entry to the main condenser.

m) Automatic recirculation of main condensate from the gland leakoff condenser outlet to main condenser will be provided by a differential pressure (across gland condenser), air pilot-operated valve.

n) The steam turbine casing drains are self-draining and run to the main condenser through a cascading system of orifices.

o) The distilling plant functions under automatic control of level and heating steam, with constant discharge of brine and salinity control of distillate. If salinity exceeds a maximum of 0.25 grains sea salt per gallon, the distillate is dumped to the bilge by means of a solenoid valve activated by the salinity control.

p) All essential motor-driven pumps are provided in duplicate, with the pump not in service arranged to start automatically on a low discharge pressure signal. An exception is the main condensate pump, the stand-by starting of which is signalled by high level in condenser drain well.

The principal electric power source in the seagoing mode is the gear-driven (attached) generator. Its output is maintained at 60 Hz (±5 Hz) over a ship speed level down to approximately 80 percent (50 percent power) by the speed/power program. Below about 80 percent ship speed in the seagoing mode, and at all times in the maneuvering mode, electric power supply is derived from the standby diesel generators.

a) The Primary Source Selector (PSS) which uses tachometer feed back to sense gear driven generator speed. When speed falls below that corresponding to 55 Hz or when the maneuvering mode is selected, the PSS signals the standby generator auto start panel to start a diesel generator, and the auto-synchronizing and load sharing panels to transfer the load. If the diesel generator fails to start or take load, the power confirmation panel (see below) restores the load to the gear driven generator.

When speed is restored to 60 Hz in the seagoing mode, the PSS signals the auto-synchronizing panel and load sharing panel which return the load to the gear-driven generator. The PSS then stops the diesel generator.

A time delay is built into the PSS to avoid unnecessary cycling in response to short term speed excursions.

b) The Power Confirmation Panel (PCP) functions to insure continuity of power supply during speed changes in the seagoing mode and at all times in the maneuvering mode. Should the diesel generators fail to start up or take load when ordered to do so by the PSS in the seagoing mode, or should both diesel generators malfunction in the maneuvering mode, the PCP activates both an alarm sequence and an emergency speed/power program. Under this program the function generator maintains shaft speed constant at a value corresponding to a gear-driven generator frequency of 60 Hz, varying only propeller pitch in response to ship speed commands. The PSS then selects the gear driven generator to supply the load. This situation remains until faults are cleared and the system is manually reset to normal operation.

c) The emergency panel functions independently of the PCP. It senses frequency and voltage, and on any wide deviation from 60 Hz or 480 volts acts through the auto start panel, the auto synchronizing panel, and the load sharing panel to start and transfer load to the selected standby generator.

d) The diesel generator auto start panel contains the diesel generator manufacturer's standard starting controls to start and bring the selected unit up to synchronous speed in response to a signal from the PSS, the emergency panel, or to a remote manual signal.

This panel contains a timer to automatically operate a motor driven pre-lube pump at intervals to maintain an oil film on the bearings of the standby diesel generators.

e) The diesel generator selection panel enables manual selection of the diesel generators for primary standby duty and secondary standby duty. It also enables one diesel to be precluded from auto start for maintenance, or both diesels to be run in parallel.

f) The auto synchronizing panel (ASP) when signalled by the PSS or the emergency panel compares frequencies on either side of the selected diesel generator's open circuit breaker and adjusts the diesel generator governor to synchronize the two. It then closes the synchronizing breaker.

g) The load sharing panel functions in either of two modes. In the "normal" mode, the load sharing panel is actuated by the ASP to transfer the load onto an oncoming generator. When this is achieved, the load sharing panel opens the circuit breaker of the unloaded generator.

In the "parallel" mode the load sharing panel functions to share the load equally between the two diesel generators.

h) The engine safety-shutdown panels contain the diesel engine manufacturer's standard protection devices. When any protective device on the panel acts to shut down a running diesel generator, the emergency panel functions to start the standby unit.

Other control devices, including governors, circuit breakers and emergency generator automatic controls, are provided in accordance with manufacturers' standard practices.

The control system is intended to permit vessel operation in all seagoing conditions as well as standby and maneuvering periods, from the bridge, without manual attendance in the machinery spaces. It is further intended that plant start-up and shut-down shall be semi-automatic, initiated at the bridge or engine room control console and involving a minimum of operator presence at the machinery. Control system operation in these normal modes for start-up, manual operation, normal shut-down, and emergency shut-down are described in the following paragraphs.

The start-up system configuration is such that the steam plant start-up and stabilization period overlaps the gas turbine start-up and maneuvering period. In normal operation, maneuvering commences and continues on the gas turbine alone, while the steam plant warms up to operating conditions. As nearly three-quarters of the rated output derives from the gas turbine, this will in no way inhibit the ship's operational flexibility. The steps during a typical starting procedure are listed below.

a) One of the two diesel generators will be in service prior to plant start-up. Critical fluid levels will have been manually checked, and operation of safety trips and other protective devices confirmed.

b) About thirty minutes prior to standby, gas turbine off-engine auxiliaries and steam turbine/reduction gear lubrication systems are started remotely. Correct operation of automatic controls is monitored at the engine room control station.

c) About fifteen minutes prior to standby, the gas turbine is started manually from the engine room control station and comes automatically to the idle condition. The steam turbine is then spinning under power of the gas turbine.

d) About five minutes after gas turbine start-up, a single push button at the engine room control station is used to initiate a sequence which starts the following equipment automatically at thirty second intervals:

- Boiler Circulating Pump
- Stack Heater Feed Water Circulating Pump
- Main Condenser Vacuum Pump
- Main Condensate Pump
- Main Condenser Circulating Pump
- Gland Leakoff Exhauster

e) The boiler pressure rise will be governed by the availability of waste heat from the gas turbine. During this period warm-up requirements are accommodated without operator attention as follows:

- When the surface temperature of the steam drum vent line reaches about 220F, a temperature sensor opens the power supply to the vent valve solenoid, allowing a spring to close the valve. This assures removal of air from the drum and permits a certain amount of deaeration of water contained in the boiler.
- Simultaneously, the boiler feed pump starts, automatically recirculating to the DFT until the drum level contol begins to close the recirculating valve and opens the feed water control valve.
- Throughout the warm-up period, the superheater is open to the main condenser, through the steam dump valve in the maneuvering mode, and through the turbine should the warm-up period overlap into the seagoing mode. Positive drainage, circulation and steam line warm-up are thus assured.

f) As the boiler stabilizes in accordance with the gas turbine load and exhaust conditions, the automatic controls function to maintain levels in the ADT, the DFT, and the boiler drum, to maintain deaerator shell pressure and gland seal pressure, to limit boiler pressure and to operate plant auxiliaries as required.

g) When the seagoing mode is selected, the turbine admission valve opens gradually, under motor control, allowing the steam dump valve to close.

h) When the plant has stabilized the distilling plant is started manually as described below.

Steam supply valve, vacuum line valve and water feed valve are manually opened spring valves. Opening each of these valves cocks a trigger-retarded closing spring.

The stop valve in the salt water line is opened to supply water to the distillate cooling nest and the evaporator shell. The shell water level is thereafter maintained automatically by a differential level, air pilot operated feed valve.

The brine pump is started.

The vacuum line to the main condenser is opened.

Heating steam stop valve is opened. Heating nest pressure is subsequently maintained automatically by a diaphragm controlled steam supply valve and orifice.

When a level accumulates in the distillate cooling section, the distillate pump is started. The salinometer will automatically dump distillate to the bilge, recirculate it to the shell, or pass it to the distilled water tank or potable water tank.

Normal operation after start-up is as described below for both the maneuvering and steady state conditions:

a) During normal operation the COGAS plant will respond to command signals from the single throttle lever on the bridge. Under steady state conditions the plant will be operating at the correct plant output power level and propeller pitch position in accordance with the command signal.

b) During transient conditions, the control system of the gas turbine can produce a change from idle to full power output in a matter of approximately 20 seconds, whereas the steam turbine cycle requires several minutes to reach equilibrium.

The actual response time of the gas turbine step changes in the throttle command signal is dictated by the torque, speed, acceleration and temperature limiting functions in the gas turbine fuel control system.

c) In the maneuvering mode, as shown in Figure 14, when the remote throttle control lever is moved from the "stop" position to a position calling for forward or astern thrust, the propeller pitch is increased to reach full pitch at approximately 30 to 40 percent throttle lever position. As the lever is moved further in the ahead or astern direction, the output power of the gas turbine is properly programmed to increase the total COGAS plant power to produce a linear increase in the steady state ship speed with the throttle lever position, to reach 100% power, ship speed and pitch at the 100% lever position.

d) In the maneuvering mode, the steam turbine admission valve is closed and the steam dump valve is opened. All power for maneuvering derives from the gas turbine alone.

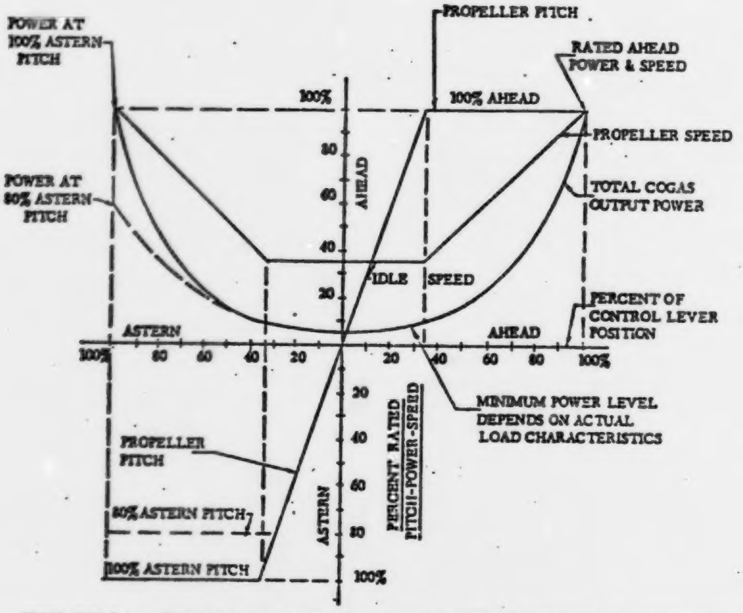


FIGURE 14 - PERFORMANCE OF COGAS SYSTEM-MANEUVERING MODE

e) In the seagoing mode, it is required to hold the shaft speed constant to maintain frequency control of the gear-driven generator. This illustrated in Figure 15. When the ship speed/direction lever is positioned in the 80% to 100% power range, the control system will adjust the propeller pitch automatically in response to a shaft speed feedback signal to hold shaft speed nominally constant.

The normal shut-down procedures are as follows:

a) The normal shut-down procedure will be to bring the throttle command lever to the "stop" position which brings the gas turbine to the "idle" condition and the propeller pitch to zero or flat pitch. A "normal stop" button is pushed to shut down the gas turbine by closing the fuel supply valves and stopping the fuel oil pump after a cool-down period of approximately 5 minutes.

b) The "normal stop" button also closes the steam turbine admission valve, opening the steam dump valve, although in most cases these steps will have occurred as a period in the maneuvering mode will have preceded the stop command. The superheater continues to vent and drain to the main condenser.

c) When the shaft is stationary, the turning gear is engaged by a solenoid and started. Gas turbine start up and steam flow to the steam turbine are prevented under this condition.

d) When DFT shell pressure falls to approximately 5 psig, a pressure switch activates a solenoid to trip the distilling plant steam supply valve and vacuum line valves. The same pressure switch also starts a timer in the distilling plant panel which stops the brine and distillate pumps after about 5 minutes, and trips closed the feed valve.

e) When steam drum pressure falls to approximately 5 psig, a pressure switch opens the drum vent and stops the boiler feed pump.

f) When evaporating bank outlet temperature is within ten degrees of its inlet temperature, the boiler circulating pump is stopped. The same signal stops the vacuum pump and the condensate pump.

g) When stack heater outlet temperature is within ten degrees of its inlet temperature, the stack heater circualting pump is stopped.

h) About ten minutes after stopping the boiler circulating pump, a timer stops the condenser circulating pump.

i) About twenty minutes after stopping the boiler circulating pump, the same timer stops the lube oil pump.

The emergency shut-down procedure may be initiated manually from a button located at each control station or automatically due to an emergency trip of the gas turbine or low lube oil pressure in the steam turbine system. The procedure differs from the normal shut-down as follows:

a) The gas turbine fuel supply valves are closed immediately. The fuel oil pumps continue to operate.

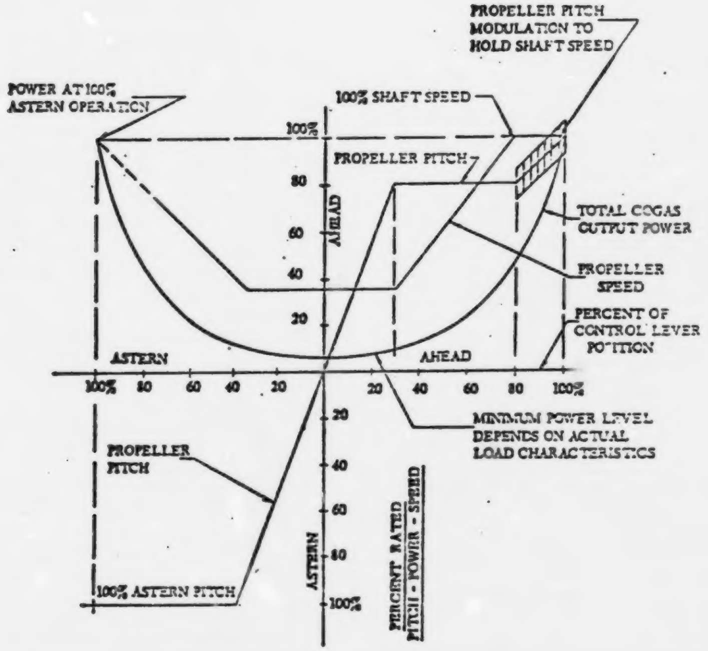


FIGURE 15 - PERFORMANCE OF COGAS SYSTEM-SEAGOING MODES

b) The steam turbine admission valve closes immediately, automatically opening the steam dump valve.

c) The speed control lever may be used to control propeller pitch, enabling the shaft to be brought rapidly to a stop if desired, by changing the propeller blades pitch.

d) If the shaft remains stopped for more than three minutes, the shaft stationary alarm sounds and within a further two minutes, unless precluded, the normal shut-down procedure is automatically invoked.

e) Before the COGAS plant can be restarted the lever must be returned to its "stop" position and the propeller pitch must be in the zero or flat pitch position.

Emergency operation of the COGAS plant by independent operation of either the gas or steam turbine is possible, thus enabling the return of the ship to port even in the event of major component casualty. Several emergency operating conditions are described below.

In the event of a gas turbine outage up to approximately 9,000 shp is available from the steam turbine. Tanker installations can use steam supplied by the auxiliary boiler for this purpose, while other installations will be provided with an emergency combustor to supply hot gases for heating the waste heat recovery boiler. The following description applies to the latter case specifically but is similar in either case:

a) If necessary, the gas turbine power output flange is manually opened at the combining gear.

b) A changeover damper is shifted manually to the emergency position, closing off the gas turbine exhaust duct and opening the emergency combustor to the boiler gas inlet.

c) In this mode, the engine room is attended. Remote manual controls are provided at the engine room console and ship speed commands are transmitted from the bridge by telephone.

e) For ahead operation, the propeller pitch is set from the engine room console at the predetermined value which optimizes steam turbine output using an independent electrical pitch control system. For maneuvering the propeller is operated at fixed pitch, and the pitch control is used only to reverse the vessel.

f) Ship speed is controlled by remote manual operation of the turbine admission valve.

g) Steam generation is controlled automatically by a steam drum pressure sensor which controls burner fuel oil supply pressure and forced draft fan inlet vane position through a mechanical linkage.

h) Electrical requirements are met by one of the diesel generators.

i) All steam plant automatic controls for boiler over pressure, feed water, etc., operate normally.

In the event of steam plant outage, the gas turbine may be operated alone. Plant control is essentially normal with the engine room unattended and ship speed controlled from the bridge through the function generator. Several points should be noted, however.

a) The gas turbine exhaust gases continue to flow through the boiler. If the mode of steam plant failure permits the boiler circulating pump, the steam dump valve, the condenser, condensate pump, condenser circulating pump and boiler feed pump to be operated, the superheater tubes will not overheat regardless of gas turbine exhaust temperature. The gas turbine may be operated up to its maximum overload rating.

b) If circulation cannot be maintained, the water in the boiler can be permitted to be boiled out or to be drained and the boiler may be operated dry. It will be necessary, however, to limit gas turbine exhaust temperature. Depending on boiler tube materials, and ambient conditions, an upper limit will have to be placed on the gas turbine firing in this mode.

c) While bridge control is used in this mode,the relation between engine and propeller characteristics is different than in the normal condition, requiring the function generator to operate with a modified speed power program. The mode selection switch is used to select this program.

ECONOMICAL RANGE OF POWER OUTPUT

The overall fuel rates at various power outputs for the COGAS power plant have been plotted and are shown in Figures 16 and 17. Sample computer output sheets for the calculations of the fuel rates are included in Appendix B. These output sheets are of interest since they give details as to the flows, temperatures, pressures, electric loads and horsepowers and similar type information for the various components of the cycle.

It will be noted from Figure 16 that the fuel rate for the power plant, for either a tanker or containership, is fairly flat between 20,000 and 30,000 shaft horsepower. This fuel rate is considerably flatter than would be achieved by operating the gas turbine alone, as noted in Figure 18. For reduced power operation the fuel rates are also quite good; for example, the overall fuel rate at 10,000 shp (one-third of rated power) is about 0.43 lbs/shp-hr. For further details see Figure 17.

From inspection and comparison with other types of power plants it can be seen that the COGAS power plant is significantly more energy-efficient, by approximately 30% to 35%, when compared to a steam power plant that is used today for a containership or a tanker. It will also be noted that the COGAS plant is more energy-efficient than an open or regenerative cycle gas turbine, any steam plant including reheat steam, or any diesel propulsion system available today.

COGAS POWER PLANT

OVERALL FUEL RATE VS. TOTAL POWER OUTPUT

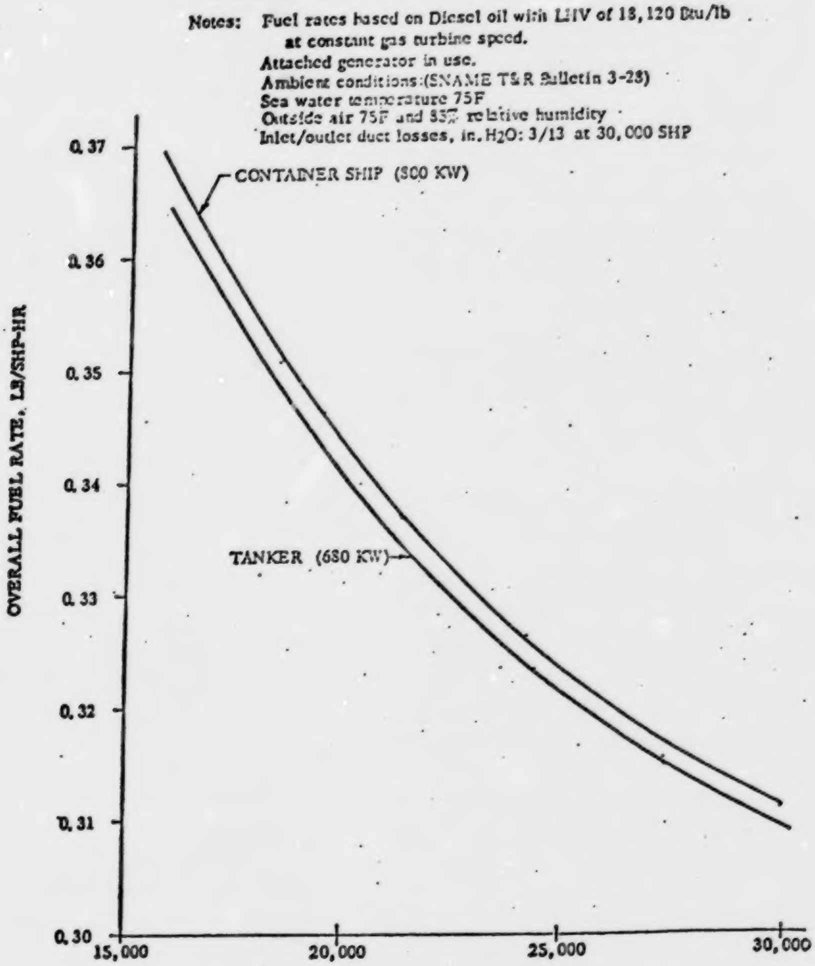

Notes: Fuel rates based on Diesel oil with LHV of 18,120 Btu/lb at constant gas turbine speed.
Attached generator in use.
Ambient conditions:(SNAME T&R Bulletin 3-28)
Sea water temperature 75F
Outside air 75F and 85% relative humidity
Inlet/outlet duct losses, in. H_2O: 3/13 at 30,000 SHP

FIGURE 16

As presented in Appendix A, some improvement in fuel consumption can be made with a COGAS plant if a multi-pressure heat recovery boiler is provided. This improvement would be in the range of 2½% to possibly 3%. However, it is believed that the additional complication does not justify the use at this time of a multi-pressure boiler for a marine service.

General Electric under contract to the U.S. Navy has a program for reducing maintenance cost and extending the life between overhauls for the LM-2500 by the use of improved gas turbine hot section materials and coatings or, alternatively, maintenance costs and life can be held constant while the gas turbine firing temperatures are increased, resulting in higher gas turbine output and improved performance. Information received from General Electric indicates that if the power turbine total inlet temperatures is increased from 1410F to 1432F, the gas turbine horsepower output is increased by approximately 5%, and the specific fuel consumption is reduced by approximately 1%. Since the start of Sharp's contract with the goverment these improvements have already been made so that the rating of the gas turbine engine of 22,200 shp (22,600 shp) selected at the beginning of the study can be increased to about 23,300 shp (23,800 bhp).

As the gas turbine improves in performance, the exhaust gas variables that affect the steam side of the COGAS plant also improve. For example, the exhaust gas temperature increases by 0.9%, the gas flow increases by 2.2%, and the specific heat of the exhaust gas increases 0.2%. The estimated improve-

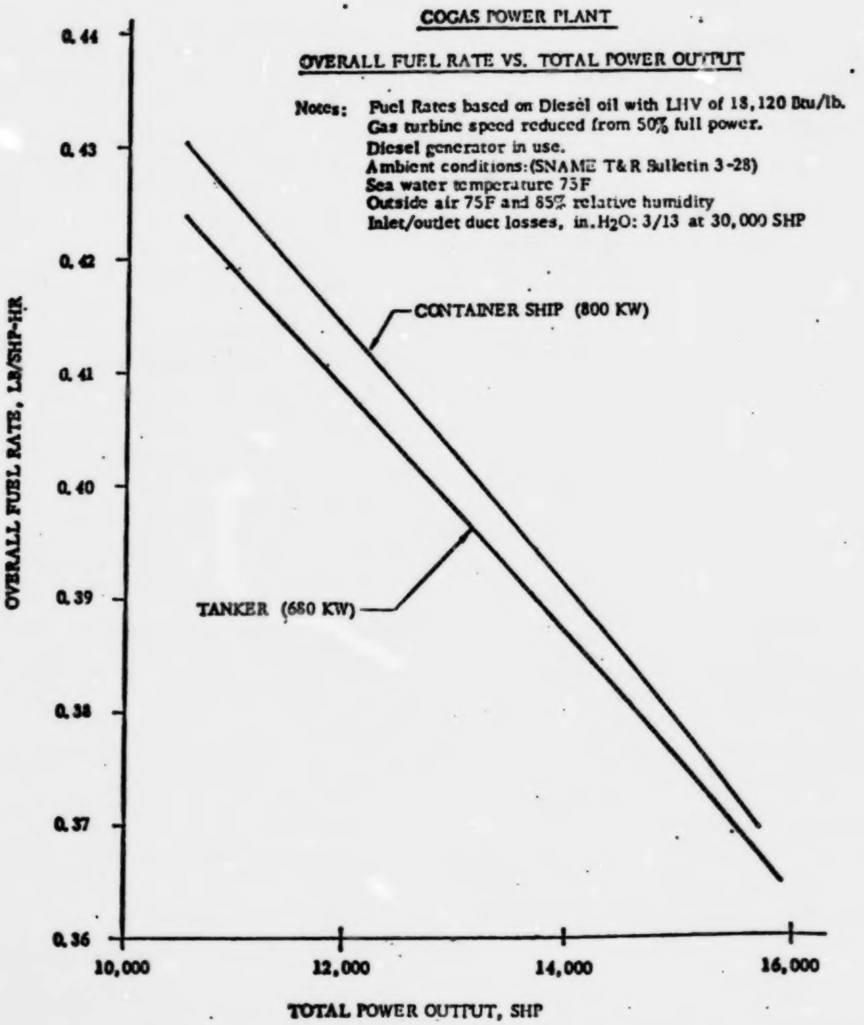


FIGURE 17

ment in the COGAS plant is a 5.5% increase in total shaft horsepower output and a 1.4% decrease in the overall fuel rate. Thus the COGAS shaft horsepower would be raised from 30,000 shp to about 31,800 shp without increasing the maintenance cost or the time between hot end section inspection or repair. Similar incremental rating increases and improvement in performance can be expected in the immediate future. This has certainly been the philosophy and practice of the aircraft gas turbine manufacturers.

If water is injected into the gas turbine along with the fuel, this will increase the gas turbine power output provided the steam firing temperature is maintained. Since firing temperature relates to gas turbine hot section life and maintenance costs, increased power output can be obtained without any significant increase in maintenance cost by the injection of water.

The data received from General Electric on water injection performance indicates an increase in gas turbine shaft horsepower, specific fuel consumption, gas flow, and specific heat while the exhaust gas temperature decreases. This data also reflects an increased firing temperature, as mentioned above, compared to the performance without water injection at the 23,300 shp rating.

The estimated performance of the COGAS power plant for increased firing temperature and water injection to fuel flow ratios of 0.2, 0.6, 1.0 indicate an increase in total shaft horsepower of 9.2, 16.0 and 20.9% respectively. The overall fuel rate will decrease 0.8% for 20% water injection rate and will increase 0.7% and 3.1% for the 60% and 100% water injection rate. These estimates include allowances for the increased steam flow to the distilling plant to accommodate the water injection. The performance with the maximum 1.0 water injection ratio is shown in Table 5.

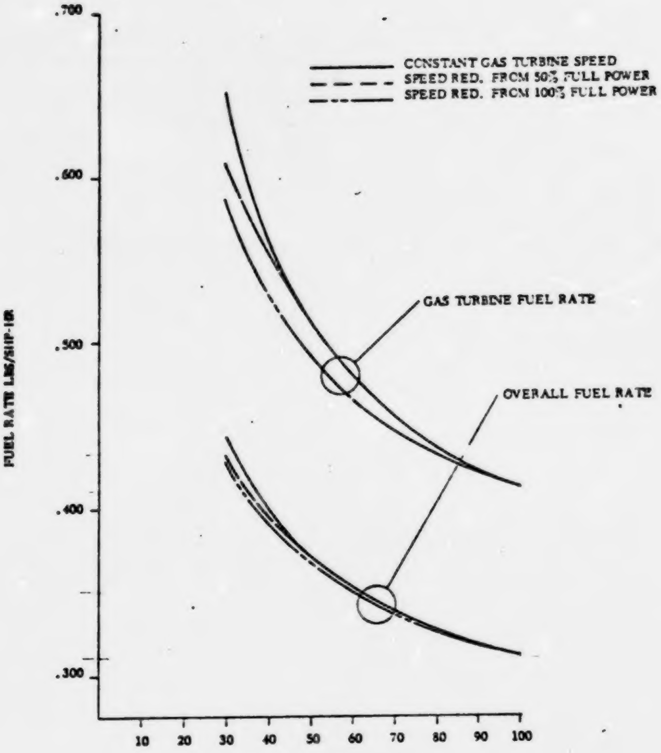


FIGURE 18 - FUEL RATES VS. GAS TURBINE POWER

The use of water injection shoreside in power plants, in addition to increasing the power output, also reduces the nitrous oxide emission which is beneficial from an environmental standpoint. It is understood that one large shoreside power plant using an aircraft derivative engine is already operating satisfactorily with water injection equal to 50% of the fuel burned.

Since the fuel rate curve with a COGAS power plant is fairly flat, between 20,000 and 31,800 shp, and further since water injection may be used which would increase the power output of the plant to about 36,700 shp, it means that the same identical power plant with all components basically the same can be used for ships requiring maximum shaft horsepower ratings between 20,000 and 36,700 shp. This will allow for a maximum of standardization, far better than can be achieved with any other gas turbine, steam turbine or diesel power plant.

It is considered that this standardization feature should be very attractive not only to commercial shipowners but to our own U.S. Navy as well.

TABLE 5

COGAS POWER PLANT PERFORMANCE
WITH WATER INJECTION

Gas Turbine Maximum Continuous Rating, SHP	27,540
Gas Turbine, BHP	28,100
Inlet Air Temp., F	75
Relative Humidity, %	85
Inlet Loss, Water Inches	3.5
Outlet Loss, Water Inches	15.3
Gas Turbine Inlet Temp., F	1,432
Gas Turbine Exhaust Temp., F	954
Fuel to Water Ratio	1 to 1
Fuel or Water Consumption, lbs/hr.	11,603
Fuel Rate Gas Turbine, lbs/bhp-hr.	0.4129
Steam Turbine, BHP	10,038
Generator Load, KW	473
Evaporator, GPD	36,000
Stack Temp., F	265
Total, SHP	36,700
Overall Fue Rate, lbs/shp-hr.	0.3162

FUEL TYPES AND PRICES

The type of fuel burned in the COGAS power plant described herein must meet the gas turbine manufacturer's specifications which, in general, are as noted in Table 6.

TABLE 6

TYPES OF FUEL BURNED

Specification No.	Title
MIL-T-5624	Grades JP4, JP5 (NATO F-40, F-44)
ASTM D975-74	Diesel Fuel Oil, Grades 1-D and 2-D
ASTM D1655 75	Turbine Fuels
MIL-F-16884	Fuel Oil - Diesel Marine (NATO F-75, F-76)
VV-F-800A	Fuel Oil - Diesel, Grades DF-A, DF-1 and DF-2 (NATO F-54)
ASTM D2880 76	Gas Turbine Fuel Oils, Grades No. O-GT, No. 1-GT, No. 2-GT, No. 3-GT
ASTM D396-76	Grades 1, 2, and 4

In general, the oil delivered to the engine shall be a minimum of 0.5 centistokes and shall be 6.0 centistokes maximum for starting and 12 centistokes maximum during operation. To meet these viscosity requirements the fuel may be heated.

Generally the maximum contaminants and other characteristics of the fuel should be as noted in Table 7 for a gas turbine engine fitted with marine coatings.

TABLE 7

MAXIMUM CONTAMINANTS IN FUEL

Ash, maximum percent	0.01
Sulfur, maximum percent	1.3
Vanadium, maximum ppm	0.5
Sodium + Potassium, maximum ppm	1.0
Lead, maximum ppm	1.0
Calcium, maximum ppm	2.0
Hydrogen content, minimum percent	12.5
Demulsification, minutes, maximum	20.0
Carbon residue, maximum (100% sample)	1.0

The commercial type of fuel available for the LM-2500 engine is a marine gas oil, meeting the ASTM D-975 specification for the No. 2-D fuel. In addition, marine diesel oil, meeting ASTM D-975 specifications for No.4-D fuel, could be used provided it is clean, that is, delivered in clean barges which are used for transport of distillate products only. A brief comparison of fuel costs per year for these two different types of fuel is noted in Table 8. It will be seen that there is considerable incentive for burning the No. 4-D fuel.

With regard to future capability of burning less expensive fuels, it is interesting to note that the Marine and Industrial Projects Department, General Electric, has undertaken a development program for LM-2500 marine gas turbines to operate on blends of residual fuel for commercial ship applications.

Elements of this program include:

- o Development of methods of inhibiting the corrosive effects of vanadium.
- o Selection of inhibiting method and additive affording suitable inhibiting or elimination of alkali metals in the fuel.
- o Development of improved methods to achieve proper combustion and long parts life.
- o Simplifying and minimizing cleaning requirements which may result from fuel additives.
- o Demonstrate LM-2500 gas turbine engine heavy fuels burning capability in 1980 land based test.

TABLE 8

#2 MARINE GAS OIL VS
#4 (CLEAN) MARINE DIESEL OIL

	#2 Marine Gas Oil	#4 Marine Diesel Oil
Low Heating Value, btu/lb.	18,120	17,590
SHP	30,000	30,000
Specific Gravity Range	0.83-0.86	0.86-0.90
Avg. Specific Gravity	0.845	0.880
Fuel Rate at 30,000 SHP	0.309	0.3183
Fuel, lbs/hr.	9,270	9,549
Fuel, bbl/hr.	31.40	31.05
Fuel Cost $/bbl **	17.44	15.63
Fuel Cost per day, $	13,142	11,647
Cost, $1000/year 350 days, 80% at sea	3,680	3,261

**New York (less barge charges and less premium for clean oil, June 1978)

It is the goal of the program to be operational on fuels at least IF-60 viscosity with contaminants typical of that blend of fuel. Table 9 gives the comparative costs of burning blended fuel of the IF-30 and IF-60 type. It will be noted that the heavier the blend, that is, the more Bunker C in the fuel, the

TABLE 9

PERFORMANCE COMPARISON
BLENDED FUELS

	IF 30	IF 60
Viscosity Redwood #1	195	430
Viscosity SSU at 100F	220	470
Blend could be		
Bunker "C"	48	74
MDO	52	26
Low Heating Value, btu/lb.	17,537	17,508
Specific Gravity	0.904	0.938
Fuel Rate at 30,000 SHP	0.3193	0.3198
Fuel, lbs/hr.	9,579	9,594
Fuel, bbl/hr.	30.32	29.28
Fuel Cost $/bbl*	12.74	12.57
Fuel Cost per day, $	9,270	8,833
Cost, $1000/year (350 days, 80% at sea)	2,596	2,473

* New York (less barge charges, June 1978)

lower the cost. It will be further noted that the cost per year for fuel is considerably less when burning blended fuel than distillate fuel. The IF-60 fuel listed is a slightly heavier blended fuel than is apparently presently being burned on aircraft derivative engines aboard the EUROLINER class ships. Thus it appears probable that General Electric should be successful in burning this blended fuel provided, of course, the fuel is properly water-washed and treated.

Although the COGAS power plant is very energy-efficient, overall yearly operating fuel costs are not significantly less than for steamships operated today and certainly more than for efficient steam power plants using reheat or diesel ships employing low-speed turning propellers and burning heavy blended oils. This problem develops since the aircraft derivative gas turbine in its present state-of-the-art must burn a distillate fuel, while steamships can burn heavy residual oil and slow turning cross head type diesel engines can burn heavy blended oil and in some cases residual oil if the contaminants and viscosities levels are satisfactory to assure reasonable engine maintenance.

Over the past few years the differential between heavy marine bunker oil and marine distillates had been widening. This is due to the fact that a great portion of the crude oil distillate is produced in unsophisticated refineries (type 2 plants) with the result that over half of each barrel of crude oil is turned into residual fuel whereas the more sophisticated refineries (type 1 plants) deliver over 80% of a barrel of crude as distillate. This large supply of residual has a depressing effect on the price compared with distillates. Unless this country and a portion of the rest of the world gets serious about saving energy from liquid petroleum products, it is probable that the current differential in price between residual and distillate will remain or possibly become greater. Figure 19 gives a comparison between the cost of marine gas oil and marine bunker oil over the past few years based upon the price in the Port of New York.

ARRANGEMENTS OF MACHINERY

Arrangement of machinery drawings were prepared for a COGAS propulsion power plant installed in a containership, see Figures 20 to 28.

The arrangements were prepared to establish the space requirements for this type of power plant. The layout of the COGAS machinery resulted in an engine room of the same length as that required by the steam power plant.

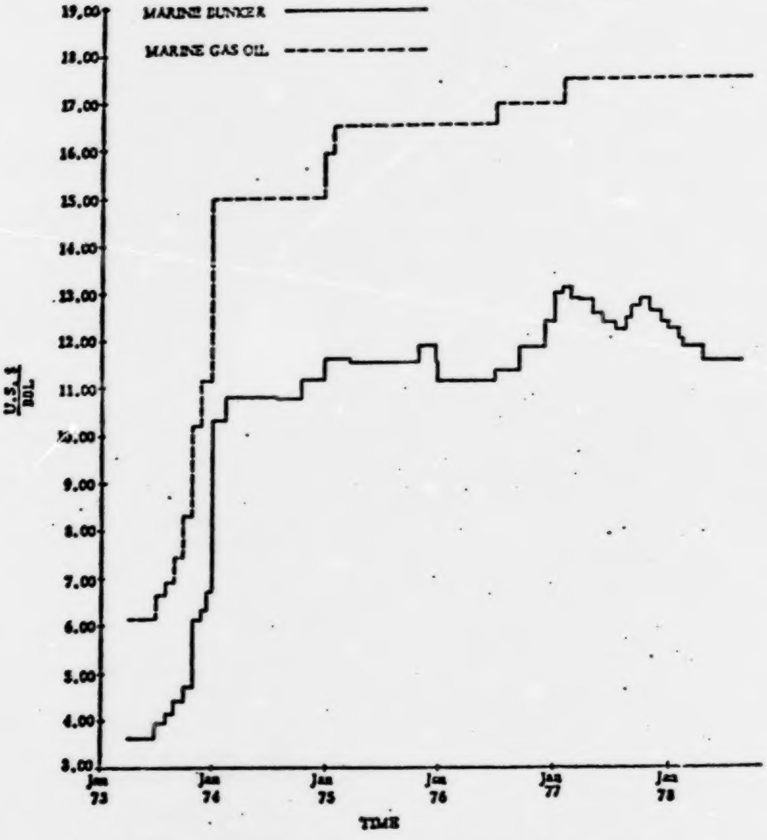


FIGURE 19 - COST OF MARINE FUEL OIL $ /BBL (1 BBL = 42 U.S.GAL.)

SOURCE: EXXON MARINE FUEL CONTRACT PRICE LIST PORT OF NEW YORK

LIST OF MACHINERY

ITEM NO.	NO. REQD.	ITEM
1	3	MACHINERY SPACE SUPPLY FAN
2	2	MACHINERY SPACE EXHAUST FAN
3	1	GAS TURBINE
4	1	STEAM TURBINE
5	1	1ST REDUCTION AND COMBINING GEAR
6	1	PLANETARY 2ND STAGE REDUCTION GEAR
7	1	INLET DUCT - GAS TURBINE
8	1	INLET DEMISTER - GAS TURBINE
9	1	EXHAUST DUCT - GAS TURBINE
10	1	ENGINE ROOM CONSOLE
11	1	MACHINERY REMOVAL AREA
12	3	INTERMEDIATE SPEED SHAFTING
13	2	SLOW SPEED SHAFTING LINE SHAFT
14	4	INTERMEDIATE SHAFT BEARING
15	2	SLOW SPEED SHAFT BEARING
16	1	LUBE OIL FILTER COALESCER - STEAM TURBINE, COMBINING & PLANETARY GEAR
17	1	LUBE OIL STORAGE TANK - DIESEL GENERATOR
18	1	LUBE OIL STORAGE TANK - STEAM TURBINE, COMBINING & PLANETARY GEAR
19	1	LUBE OIL SETTLING TANK - STEAM TURBINE, COMBINING & PLANETARY GEAR
20	1	LUBE OIL LOW HEAD GRAVITY TANK - STEAM TURBINE, COMBINING & PLANETARY GEAR
21	1	LUBE OIL SUMP TANK - STEAM TURBINE, COMBINING & PLANETARY GEAR
22	1	LUBE OIL STORAGE AND CONDITIONING ASSEMBLY - GAS TURBINE
23	1	LUBE OIL COOLER - STEAM TURBINE, COMBINING & PLANETARY GEAR
24	1	SLUDGE TANK
25	1	USED LUBE OIL TANK - DIESEL GENERATOR
26	1	MAIN CONDENSER
27	2	VACUUM PUMP - MAIN CONDENSER
28	1	GLAND LEAKOFF CONDENSER
29	1	GLAND LEAKOFF EXHAUSTER
30	1	ATMOSPHERIC DRAIN TANK
31	1	MAIN CONDENSER CIRCULATING PUMP
32	2	S.W. CIRCULATING BOOSTER PUMP - DIESEL GENERATOR
33	2	SALT WATER SERVICE PUMP
34	1	FIRE PUMP
35	1	FIRE AND TANK WASHING PUMP
36	1	BILGE PUMP
37	1	BILGE, BALL. AND S.W. SERVICE PUMP
38	2	PRIMING PUMPS AND VACUUM TANK - BILGE & BALLAST
39	2	MAIN CONDENSATE PUMP
40	2	MAIN FEED PUMP
41	2	STACK HEATER FEED WATER CIRCULATING PUMP
42	1	E.R. BILGE WATER SEPARATOR AND AUTOMATIC BILGE PUMP
43	2	JACKET WATER STANDBY CIRCULATING PUMP - DIESEL GENERATOR
44	3	WORK BENCH
45	2	LUBE OIL SERVICE PUMP - STEAM TURBINE, COMBINING & PLANETARY GEAR
46	2	POST LUBE OIL PUMP - DIESEL GENERATOR
47	2	POTABLE WATER PUMP AND COMPRESSION TANK
48	2	FUEL OIL TRANSFER PUMP -PROPULSION
49	1	TANK WASHING HEATER
50	2	GAS TURBINE FUEL SERVICE BOOSTER PUMP
51	1	BILGE AND BALLAST MANIFOLDS
52	1	F.O. TRANSFER MANIFOLDS
53	1	FUEL OIL TRANSFER PUMP -SHIP'S SERVICE
54	2	POTABLE WATER TANKS
55	2	POTABLE WATER PURIFIER
56	1	SCUTTLE BUTT AND PUMP
57	1	HOT WATER HEATER AND PUMP
58	1	TANK WASHING PUMP
59	2	BOILER CIRCULATING PUMP
60	1	MARINE SANITATION DEVICE
61	2	STARTING AIR COMPRESSOR - DIESEL GENERATOR
62	2	MAIN AIR COMPRESSOR
63	1	CONTROL AIR COMPRESSOR
64	1	SHIP'S SERVICE AIR TANK
65	1	CONTROL AIR TANK
66	2	STARTING AIR TANK - DIESEL GENERATOR
67	2	REFRIG. FILTER - SHIP'S SERVICE
68	1	REFRIG. FILTER - CONTROL AIR
69	1	HEAT RECOVERY BOILER
70	2	FUEL OIL CENTRIFUGAL PURIFIER
71	2	FUEL OIL SERVICE TANK - GAS TURBINE
72	2	FUEL OIL SERVICE TANK - DIESEL GENERATOR
73	1	FUEL OIL DRIP TANK - DIESEL GENERATOR
74	2	FUEL OIL PREFILTER
75	2	FUEL OIL FILTER SEPARATOR
76	4	FUEL OIL HEATER
77	1	DEAERATING FEED TANK
78	2	DISTILLED WATER TANKS
79	1	FEED TESTING EQUIPMENT
80	1	DISTILLING PLANT
81	2	SHIP'S SERVICE REFRIG. COMPRESSOR ASSEMBLY
82	2	AIR CONDITIONING COMPRESSOR ASSEMBLY
83	1	GENERATOR - DRIVEN BY PROPULSION UNIT
84	2	DIESEL GENERATOR - WITH ATTACHED AUXILIARIES
85	2	DIESEL GENERATOR MUFFLER
86	1	MAIN SWITCHBOARD
87	9	GROUP CONTROL CENTERS
88	1	LOW SEACHEST - MAIN
89	1	HIGH SEACHEST - MAIN
90	1	LOW SEACHEST - AUXILIARY
91	1	HIGH SEACHEST - AUXILIARY
92	1	ENGINEER'S WORKSHOP
93	1	ENGINEER'S STOREROOM
94	1	ELECTRICAL WORKSHOP
95	1	EMERGENCY COMBUSTOR
96	1	TRIMMING PUMP
97	1	CARGO DEHUMIDIFICATION UNIT
98	8	CARGO REFRIG. COMPRESSOR ASSEMBLY
99	4	FREON PUMPS
100	2	CONSOLE CARGO REFRIG.
101	1	CARGO REFRIG. PUMP OUT RECEIVER
102	1	REFRIG. COMP. TRANSFER UNIT
103	2	FREON ACCUMULATOR
104	1	PORT USE AND START-UP BOILER
105	1	FORCED DRAFT FAN - EMERGENCY COMBUSTOR
106	1	PORT USE AND START-UP BOILER FUEL OIL SERVICE TANK
107	1	CP PROPELLER OIL DISTRIBUTION BOX

FIGURE 20

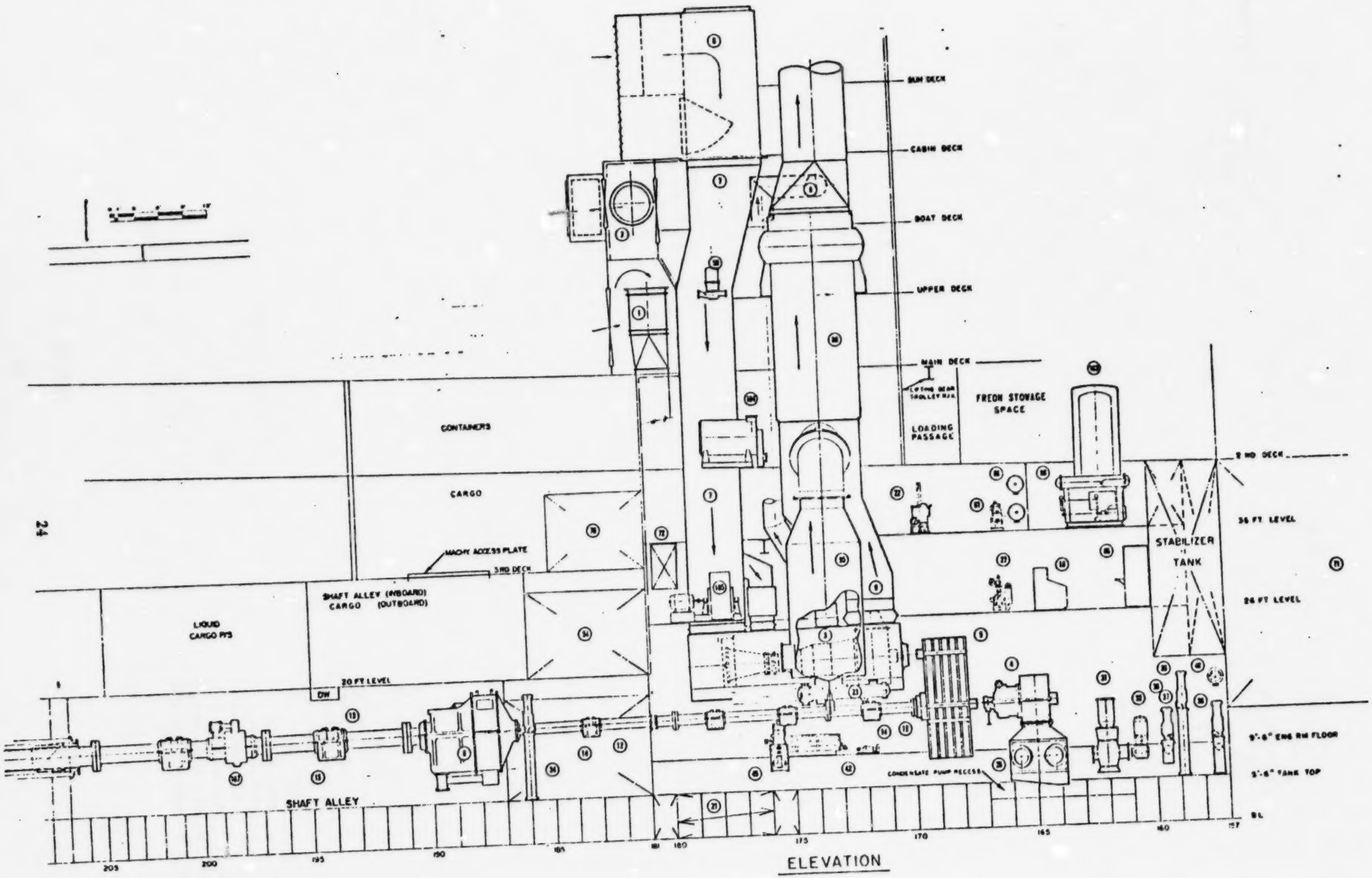
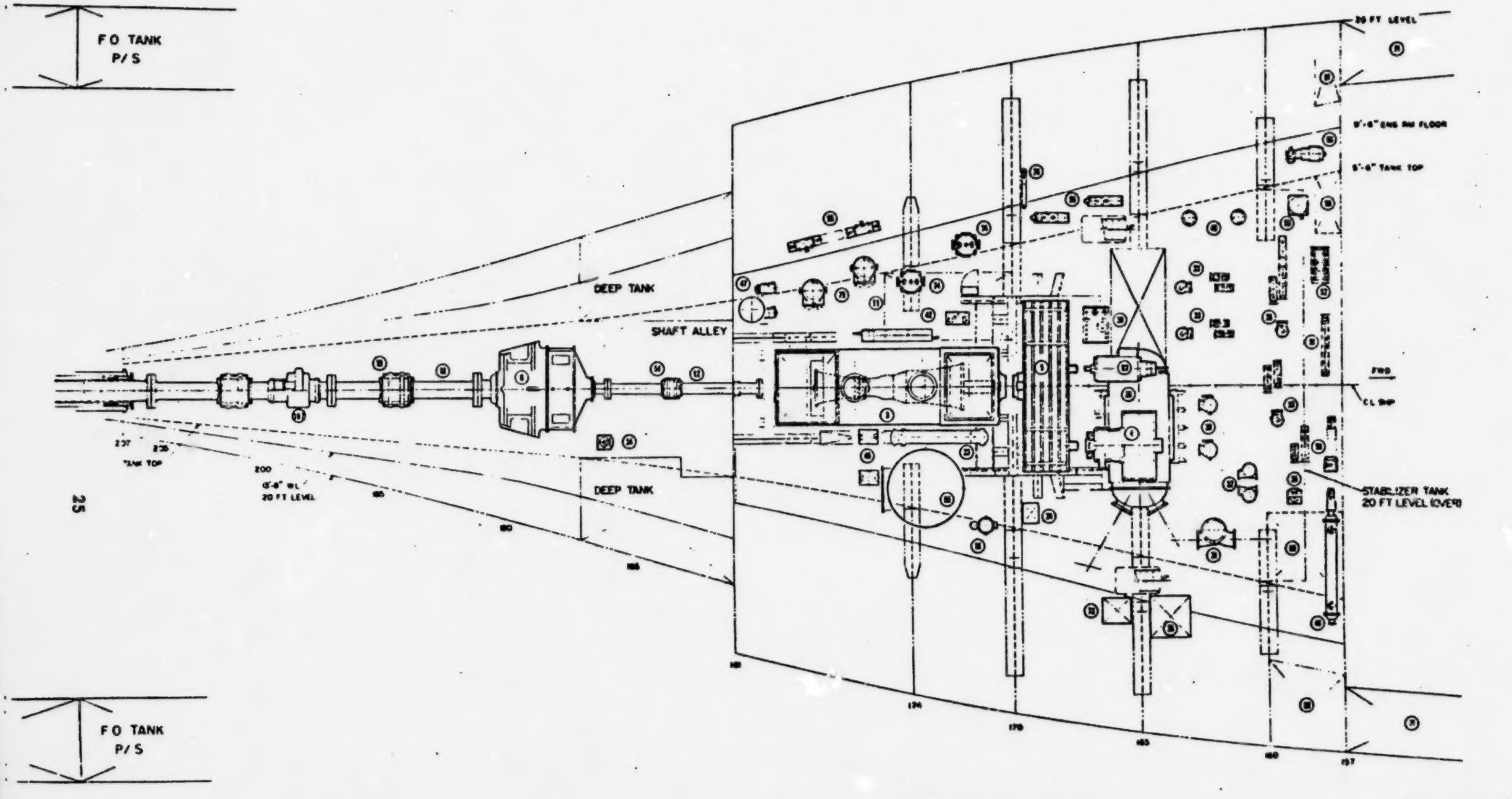
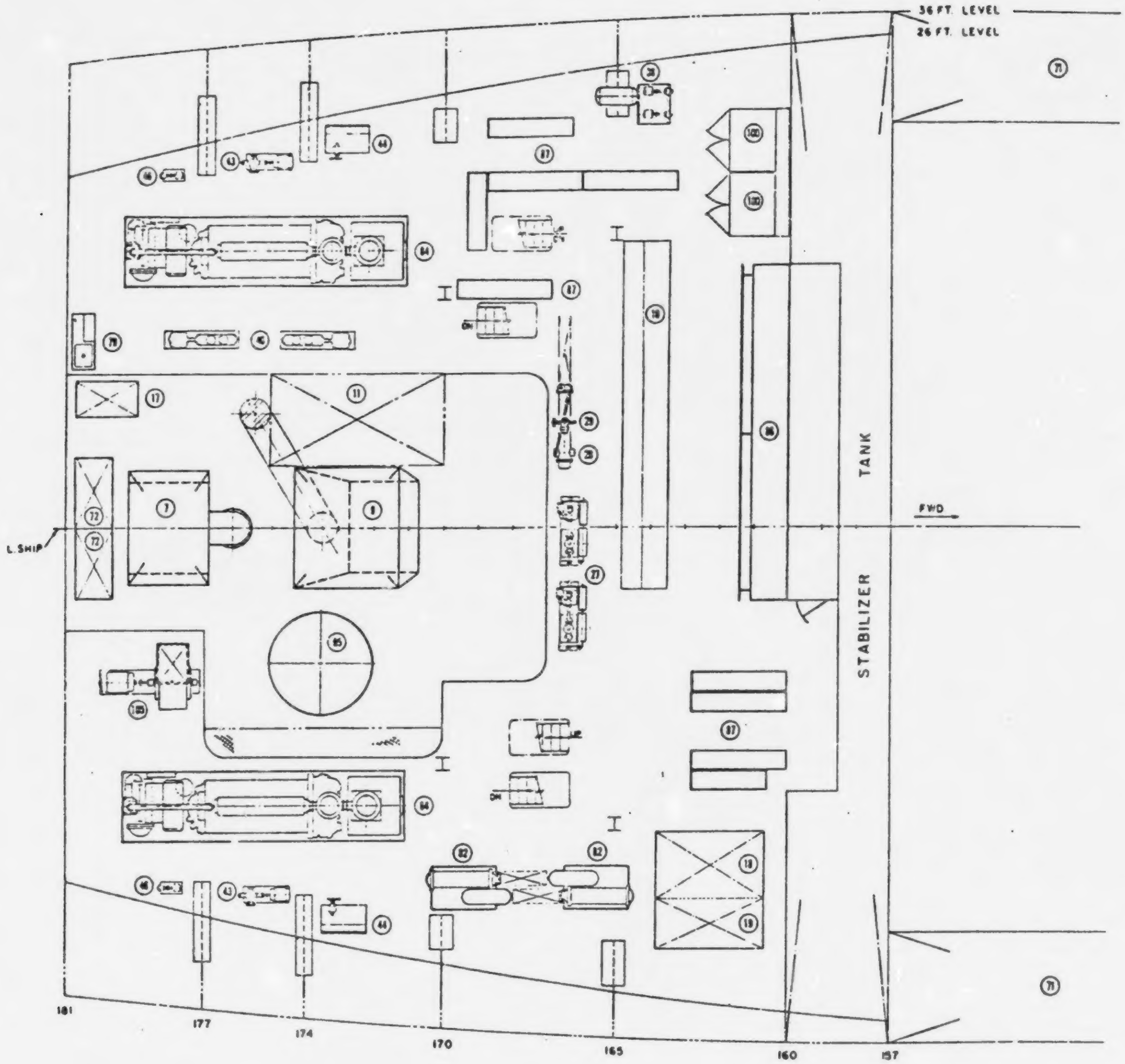

SUN DECK
CABIN DECK
BOAT DECK
UPPER DECK
MAIN DECK
LIFTING GEAR TROLLEY RAIL
LOADING PASSAGE
FREON STOWAGE SPACE
2 ND DECK
36 FT LEVEL
26 FT LEVEL
STABILIZER TANK
9'-6" ENG RM FLOOR
5'-6" TANK TOP
BL
CONTAINERS
CARGO
MACHY ACCESS PLATE
3RD DECK
SHAFT ALLEY (INBOARD) CARGO (OUTBOARD)
LIQUID CARGO P/S
20 FT LEVEL
DW
CONDENSATE PUMP RECESS
SHAFT ALLEY
205
200
195
190
185
181
180
175
170
165
160
157
ELEVATION

FIGURE 21


F O TANK
P/S
F O TANK
P/S
DEEP TANK
SHAFT ALLEY
DEEP TANK
20 FT LEVEL
5'-6" ENG RM FLOOR
5'-6" TANK TOP
FWD
CL SHIP
STABILIZER TANK
20 FT LEVEL (OVER)
TANK TOP
200
20 FT LEVEL
174
170
165
160
157

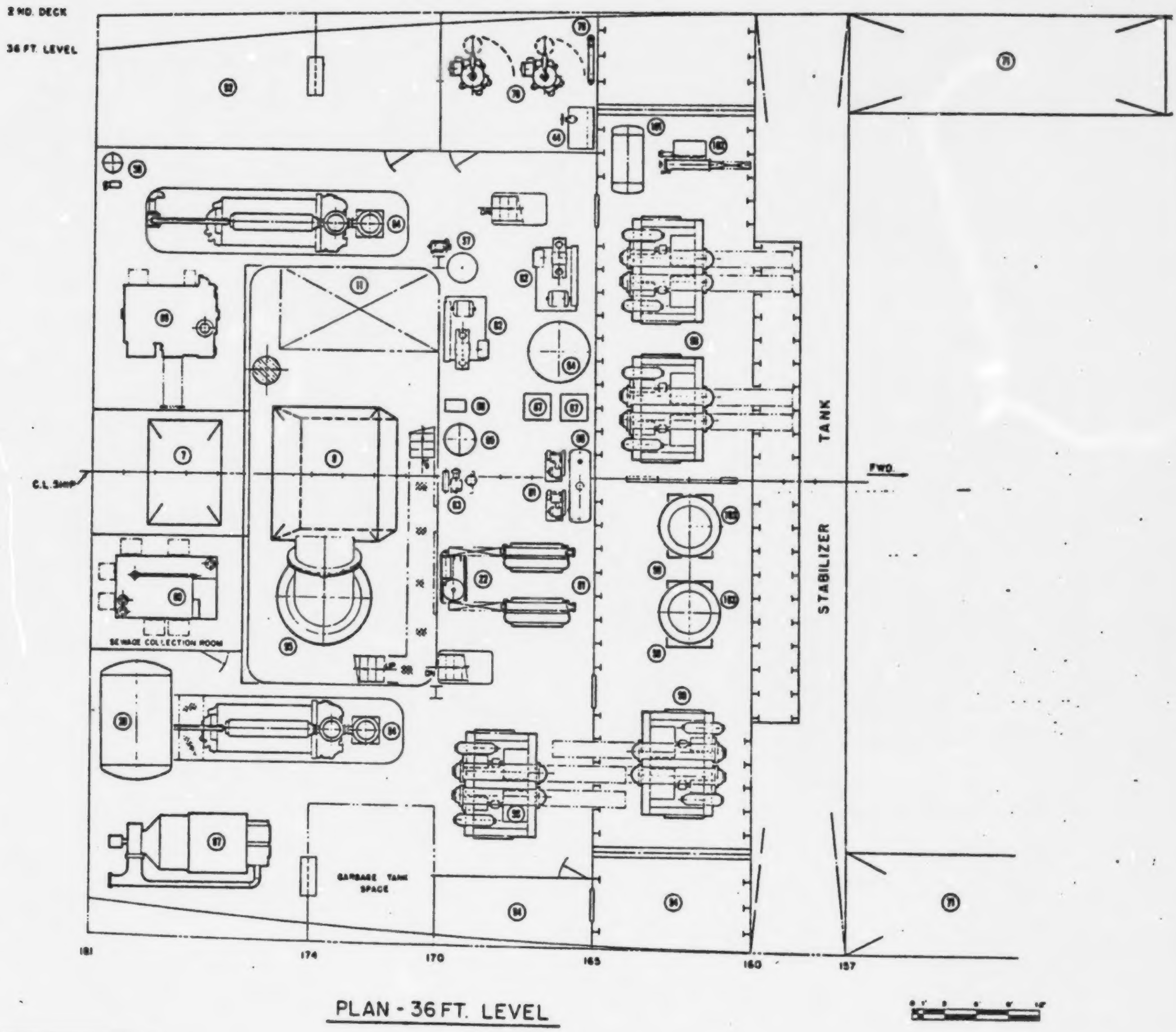
PLAN - ENGINE ROOM FLOOR

FIGURE 22


36 FT. LEVEL
26 FT. LEVEL
TANK
STABILIZER
FWD
L SHIP
181
177
174
170
165
160
157

PLAN - 26 FT. LEVEL

FIGURE 23


2 ND. DECK
36 FT. LEVEL
C.L. SHIP
FWD.
TANK
STABILIZER
SEWAGE COLLECTION ROOM
GARBAGE TANK SPACE
181
174
170
165
160
157

PLAN - 36 FT. LEVEL

FIGURE 24

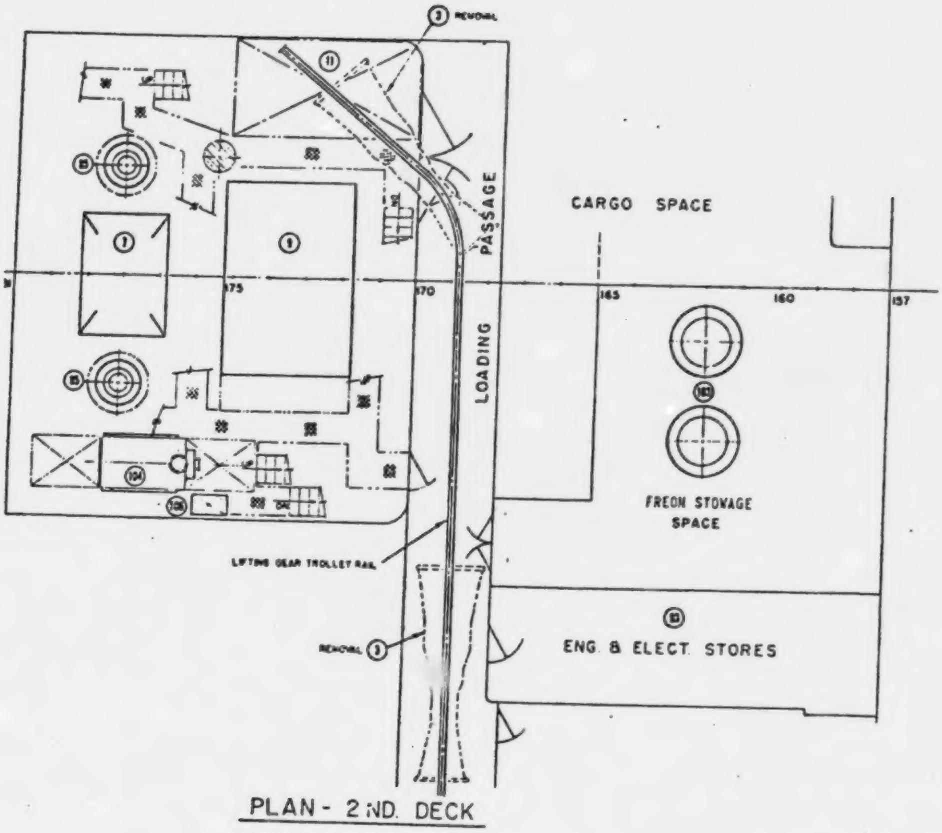
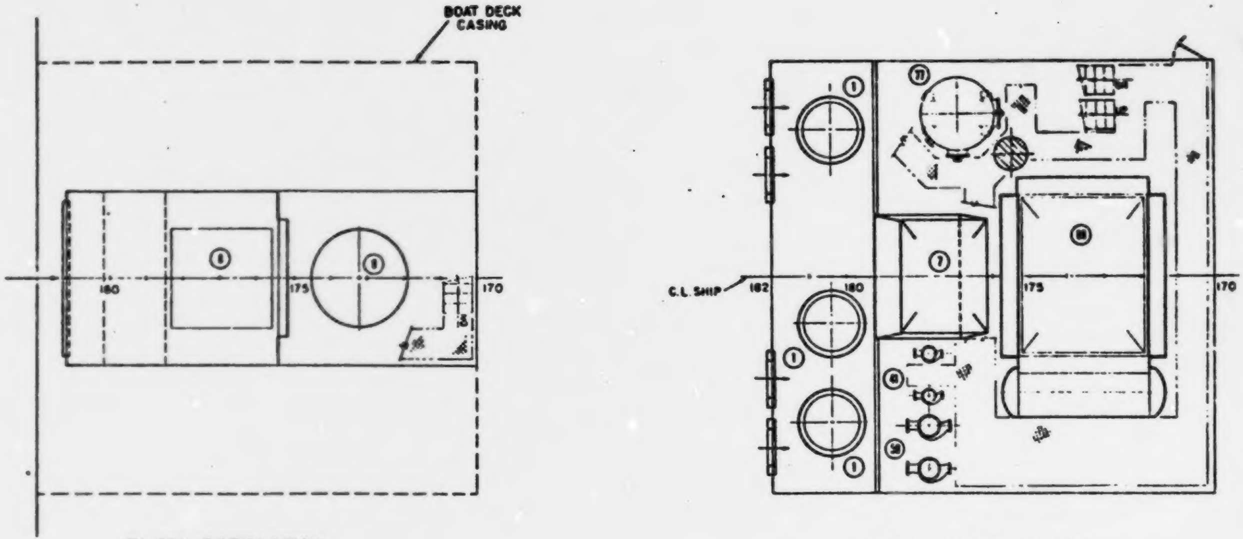
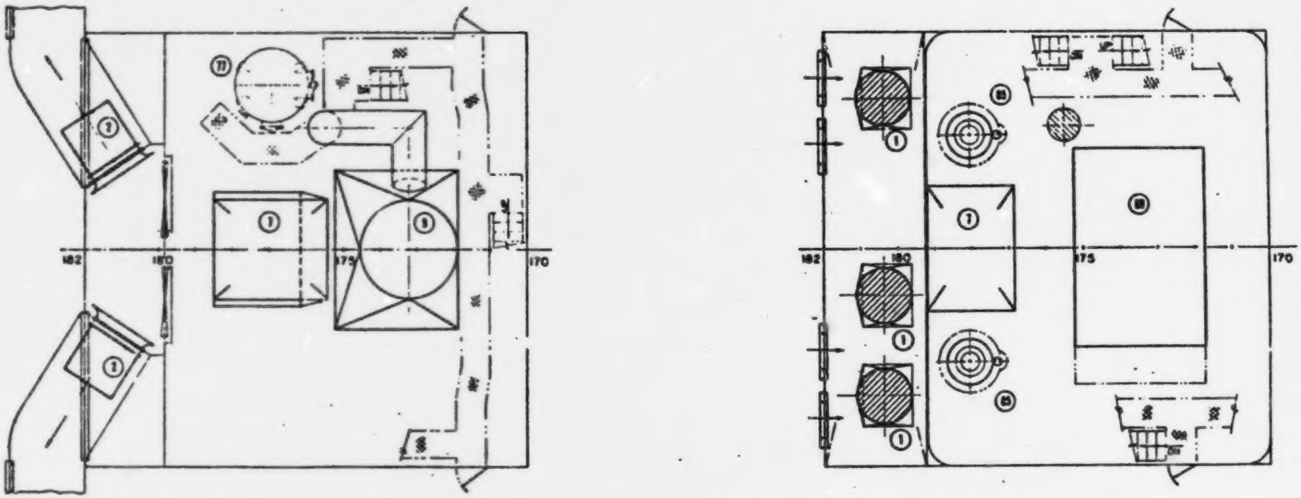

③ REMOVAL
⑪
⑦
⑨
175
170
165
160
157
PASSAGE
LOADING
CARGO SPACE
FREON STOWAGE SPACE
ENG. & ELECT. STORES
LIFTING GEAR TROLLEY RAIL
REMOVAL ③
UP
104
106
PLAN - 2ND. DECK

FIGURE 25


BOAT DECK CASING
180
175
170
C.L. SHIP
182
180
175
170

PLAN - CABIN DECK

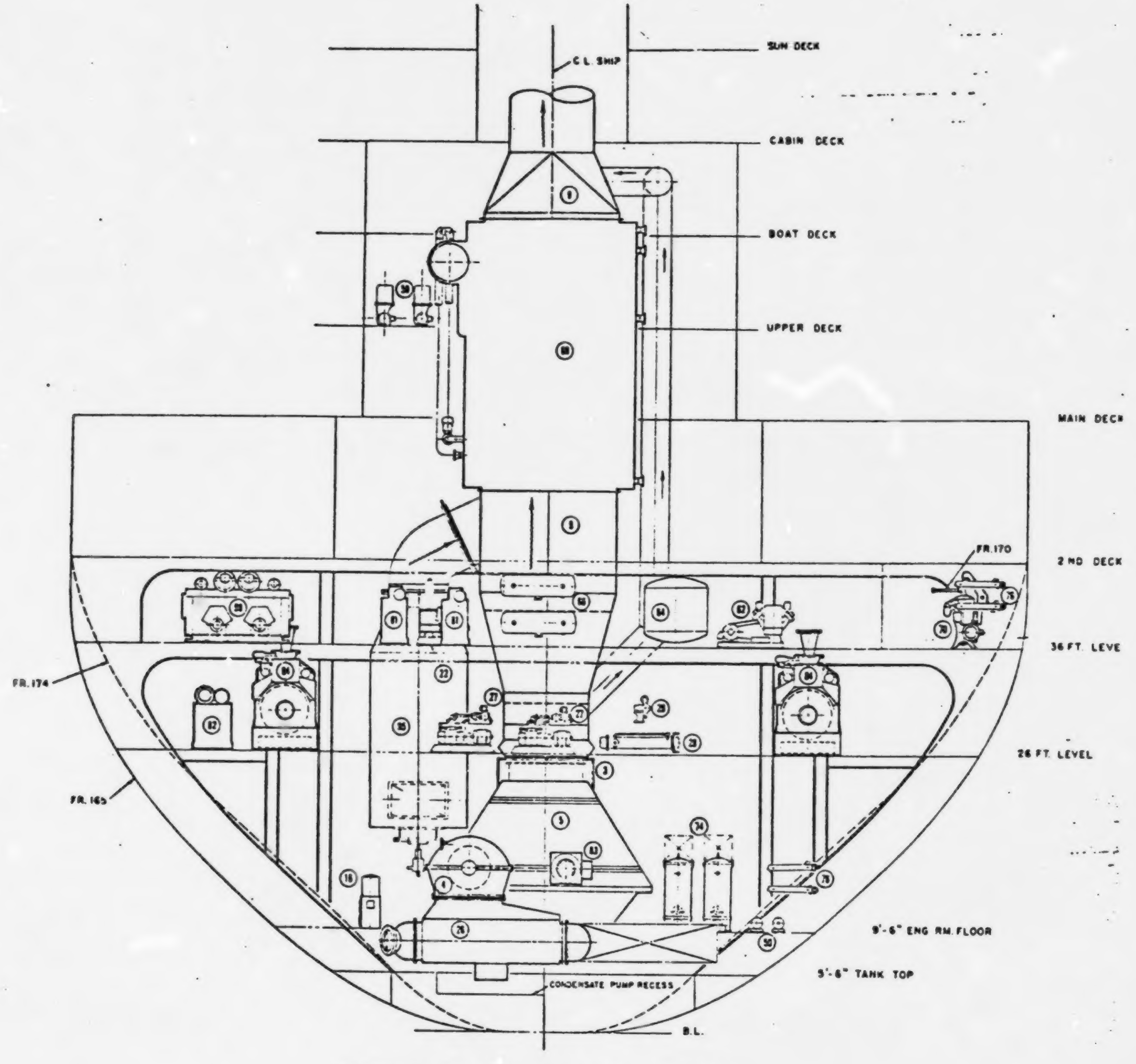
PLAN - UPPER DECK


182
180
175
170
182
180
175
170

PLAN - BOAT DECK

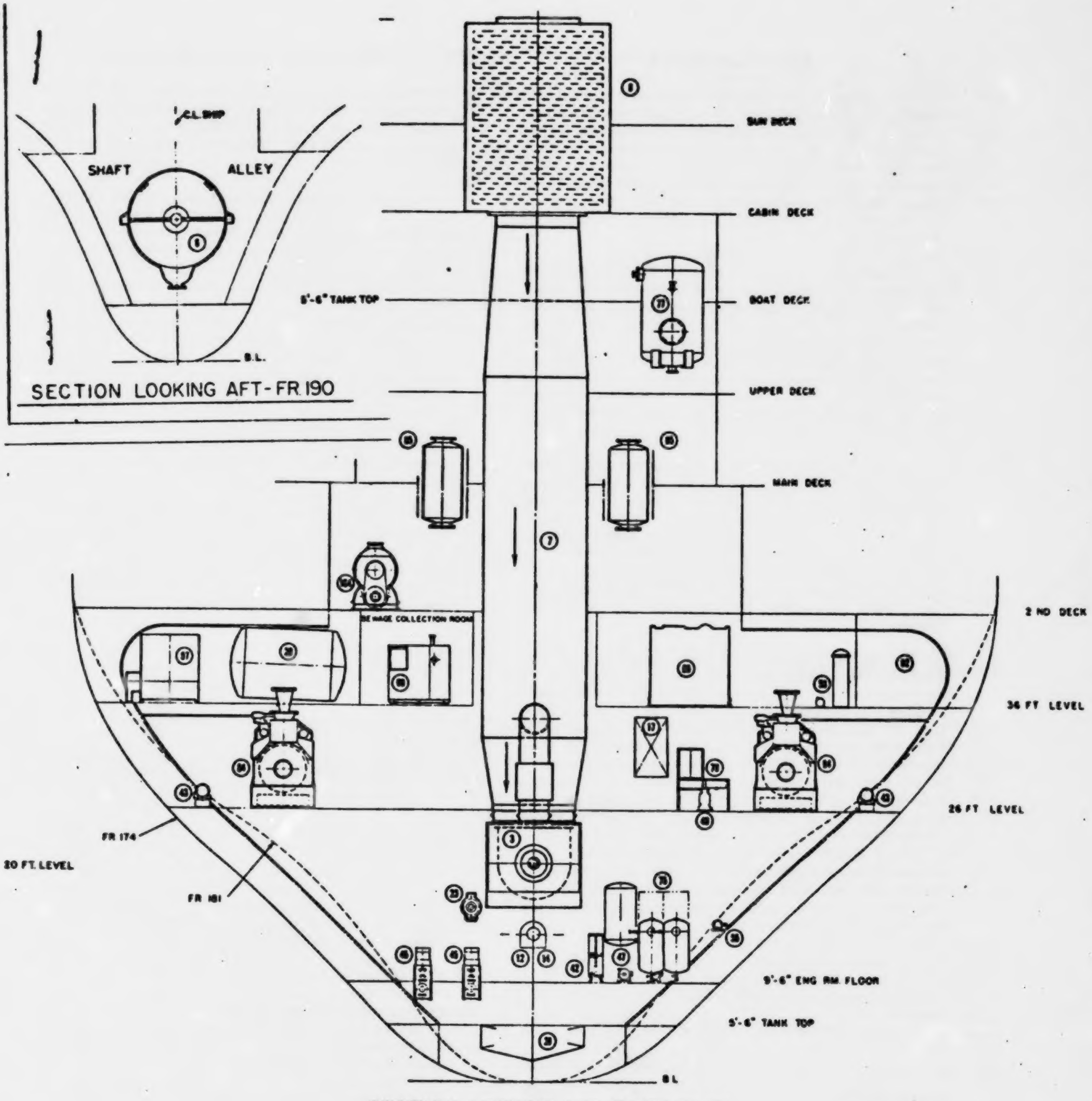
PLAN - MAIN DECK

FIGURE 26


SUN DECK
C.L. SHIP
CABIN DECK
BOAT DECK
UPPER DECK
MAIN DECK
2 ND DECK
FR.170
36 FT. LEVE
FR.174
26 FT. LEVEL
FR.165
3
5
4
22
27
74
9'-6" ENG RM. FLOOR
5'-6" TANK TOP
CONDENSATE PUMP RECESS
B.L.

SECTION LOOKING AFT - FR.165 TO 174

FIGURE 27



SECTION LOOKING AFT - FR.190

SECTION LOOKING AFT - FR.174 TO 181

C.L. SHIP
SHAFT
ALLEY
B.L.
SUN DECK
CABIN DECK
5'-6" TANK TOP
BOAT DECK
UPPER DECK
MAIN DECK
SEWAGE COLLECTION ROOM
2 ND DECK
36 FT LEVEL
26 FT LEVEL
20 FT. LEVEL
FR 174
FR 181
7
3
12
14
42
47
9'-6" ENG RM. FLOOR
5'-6" TANK TOP
B L

FIGURE 28

WEIGHT AND SPACE

The weight of the COGAS propulsion power plant is considerably less than that of a typical modern steam plant. A comparison of the machinery weights, MarAd groups 20 to 29, for a large containership with a maximum continuous power rating of 30,000 shp and a propeller speed of 105 rpm indicates that the COGAS plant weighs approximately one-third less than a corresponding steam plant. Details of the comparison are shown in Table 10.

The outfit and steel weights for the COGAS plant are also less than those of a corresponding steam plant. The outfit weight savings are mainly due to the elimination of the fuel oil heating coils in storage tanks and the reduced rating of the electric power plant. The steel weight saving is due to the reduced size of the superstructure to house the smaller crew associated with the COGAS plant. In addition, because of the COGAS plant lower fuel rate, the weight of bunker fuel oil is less.

For a containership with the same cargo deadweight, the combined effect of all the weight savings associated with a COGAS installation is to reduce the ship displacement and thus permit a decrease in shaft horsepower for maintaining the same speed over the route. For a tanker with the same displacement, the savings in machinery, outfit, steel and bunker fuel weights permit an increase in cargo deadweight and thus results in a higher earning power for the COGAS tanker.

The space required for the installation of the COGAS power plant was determined on the basis of arrangement of machinery drawings which were prepared to show the COGAS plant in a containership and a tanker. Corresponding arrangement of machinery drawings for steam installations in the same type of ships were already available and this allowed a detailed comparison of the space requirements for the two types of power plant. As a result of the evaluation, it was determined that, for the two cases studied, the COGAS and steam plants require the same length of engine room and thus essentially the same volume of machinery space. In a more general way, it can be assumed that the space requirements for the COGAS plant are the same or only marginally less than that required for a steam plant.

TABLE 10

MACHINERY WEIGHT COMPARISON

	COGAS	Steam
Main propulsion, L.Tons	192.6	258.0
Feed and condensate systems	15.8	51.6
Evaporator system	4.8	10.0
Shafting and propeller	190.0	176.0
Lube oil system	18.2	20.6
Air systems	9.8	9.8
Boilers, uptakes, and fuel oil system	148.5	287.0
Steam piping systems	13.8	48.0
Access, workshop, mach.vent., et al	80.0	94.0
Sub-total	673.5	955.0
Liquids	81.0	175.0
Total, L.Tons	754.5	1130.0

ELECTRIC GENERATOR LOADS

Generator load analyses for steam and COGAS power plants installed in a reefer containership were prepared and are tabulated in Table 11. The analyses assume operation at a maximum continuous rating of 28,500 shp.

The electric power plant of the COGAS installation consists of an attached generator and two diesel generators. At sea, the attached generator provides all electric power when the propulsion plant output varies from 100 to 50 percent of maximum continuous power rating. Below the 50 percent power level and when in port, electric power is supplied by one of the two diesel generators, the other being a stand-by. The size of the attached generator is selected on the basis of the at-sea load, and for the case analyzed above, an attached generator size of 1000 kw is considered adequate. The size of the diesel generators, on the other hand, must be selected on the basis of the maximum electric load requirements and may be larger than that of the attached generator because of special ship services load demand, such as reefer pulldown, cargo handling, etc.

The electric power plant of the steam installation consists of two ship service generators. They may be both steam turbine driven, both diesel driven, or one steam turbine and one diesel driven. The size of the generators must be selected on the basis of the maximum electric load requirements.

TABLE 11

COMPARISON OF GENERATOR LOADS
STEAM AND COGAS

	Steam	COGAS
Forced draft fans	209.8 kw	0 kw
Main feed pump	0	56.2
Main condensate pump	23.7	12.3
Main condenser circ.pump	0	39.0
Heat recovery boiler circ.pumps	0	13.1
Main condenser vacuum pump	15.9	15.9
Air preheater drive and ADT drain pump	9.5	0
F.O. system (F.O.service & trans.pumps, purifiers)	17.2	17.3
L.O. system (L.O. serv. & stern tube pumps, filter & heater)	32.5	27.6
Air compressors	9.6	5.8
Gland leakoff exhauster	1.9	1.9
G.T. enclosure fan	0	7.6
Machinery space ventilation fans	134.5	134.5
Lighting	9.5	9.5
Controls	2.0	2.4
Steering gear room equipment	7.1	7.1
Propulsion	473.2	352.2
Ship's Service	70.8	70.8
Distiller Plant	15.0	7.0
Hotel	190.0	169.0
Deck Machinery	21.0	21.0
Cargo Refrigeration	180.0	180.0
Total	950 KW	800 KW

ECONOMICS AS APPLIED TO A CONTAINERSHIP

This comparison uses a Sharp's recent design for C8 type containerships as the basis in the analysis. These ships have efficient steam propulsion plants with a maximum power output of 28,500 shp with steam conditions of 850 psig/950F at the propulsion turbine inlet, 1.5 in. HGA condenser pressure, four stages of feed heating and regenerative air heaters.

The containership with the COGAS propulsion plant is alike in all other principal respects, having the same hull form, dimensions, speed, and cargo carrying capacity. Principally because of the lower fuel consumption of the COGAS plant, however, the weight of the fuel and service displacement is less, permitting a small decrease in draft and horsepower required for maintaining the same speed over the route. The two containerships have the characteristics indicated on Table 12.

TABLE 12

SHIP CHARACTERISTICS

	Steam	COGAS
Length overall, ft. (m)	813 (248)	Same
Beam, ft. (m)	90 (27.4)	Same
Depth, ft. (m)	53 (16.2)	Same
Draft, service, ft. (m)	29 (8.84)	28.5 (8.67)
Speed, service, knots	22	Same
SHP, maximum continuous	28,500	28,150
TEU (maximum)	1,708	Same

The ships are considered to run at the maximum continuous rating of the machinery to achieve the average service speed of 22 knots.

The ships are considered to sail from Newark, N.J. to Brisbane, Australia via the Panama Canal then on to Sydney and Melbourne in Australia, returning to Newark via Wellington, New Zealand and the Panama Canal to New Orleans, Charleston and Philadelphia. At 22 knots the ships require nearly 60 days for a complete voyage of 24,000 miles, calling at eight ports for cargo. This results in the annual operating breakdown below:

Days at sea	265
Days in port	70
Days maneuvering	9
Days standing-by	6
Operating days per year	350

The COGAS and steam containerships differ in their machinery and, because of the reduced crew for the COGAS installation, in their accommodations. A careful evaluation of these differences resulted in estimated differences in capital costs for the two ships, and these in turn were converted to corresponding yearly charges.

The steam containership was assumed as the basis for the comparison since all the required information on this power plant was available, including machinery specifications, arrangement of machinery drawings, and cost of all the machinery installed in the engine room. As the comparison includes only part of the engine room equipment, i.e. the main propulsion machinery and associated auxiliaries, only the affected machinery was identified and its cost including installation, was upgraded to reflect mid-1978 prices. To determine the cost of the corresponding main propulsion machinery and associated auxiliaries for the COGAS power plant, purchase specifications for this equipment were prepared and mid-1978 prices were obtained. In addition, the cost of installing this equipment was calculated using arrangement of machinery drawings and piping diagrammatics prepared as part of the MarAd project. This evaluation was conducted by the Sun Shipbuilding and Dry Dock Company, and therefore it is based on actual shipyard estimates of installed costs of the two propulsion power plants.

As a result of the above evaluation, it was found that the net cost of labor and material for the COGAS plant increases for the following cost groups when compared to the steam propulsion plant:

Shafting, bearings and propellers
Stacks, uptakes, and air intake system
Main engine and reduction gears

The net cost of labor and material decreases for the following cost groups:

Boilers
Feed and condensate systems
Main condenser and S.W. cooling
Steam and exhaust piping and drains
Fuel oil system
Electric power generation
Central controls
Other equipment, such as forced draft fans, distiller, ratings and insulation

The net cost difference for labor and material is a $600,000 saving for the COGAS installation in a new ship construction.

As discussed later herein, the COGAS installation requires a smaller crew. This reduction of required accommodations for six men results in an estimated net cost saving for labor and material of $200,000, for a total capital cost difference between the two containerships of $800,000 in favor of the COGAS installation. Assuming that these ships qualify for a maximum construction differential subsidy of 50 percent, the after subsidy acquisition cost of the COGAS containership is $400,000 less than that of the steam containership.

In this analysis the annual cost of invested capital has been taken as 8.6% on the basis of a 25 year sale/lease back arrangement between a lending institution and a favored shipowner/operator-client at 7% interest, with tax advantages and residual value accruing to the lender. Higher costs of invested capital favor the COGAS ship. Based on this financing scheme, the yearly charges for the differences in capital costs are as follows:

	Steam	COGAS
Capital Cost, before subsidy, $1000	Base	-800
Capital Cost, after subsidy, $1000	Base	-400
Yearly Charges, $1000	Base	- 34

The specific fuel consumption for the steam containership is 0.473 lb/shp-hr at the 28,500 shaft horsepower output with a generator load of 950 kw and an evaporator output of 11,100 GPD. This value is based on heat balance calculation for the steam containership selected as the basis of comparison. A somewhat lower fuel rate could be achieved, but would require an increase in the capital cost of the steam power plant to incorporate the necessary upgrading of the plant components. The type of fuel burned in the boilers is residual fuel oil with a standard higher heating value of 18,500 Btu/lb at 100F.

The specific fuel consumption for the COGAS containership is 0.315 lb/shp-hr at the 28,150 shaft horsepower output with a generator load of 800 kw and an evaporator output of 5,000 GPD. The type of fuel burned in the gas turbine is No. 2 diesel oil (marine gas oil, MGO) with a standard lower heating value of 18,120 Btu/lb. As the LM-2500 gas turbines on the ADM. WM. M. CALLAGHAN are normally operated on No. 4 diesel oil (marine diesel oil, MDO), the fuel cost comparison is also made on the basis of this fuel. A supplementary calculation is also carried out with a cut residual fuel, IF60, a light intermediate fuel (439 Redwood No. 1 sec. at 100F). This is the heaviest of blended fuels included in the manufacturer's program to determine the operational capability of the LM-2500 engines on lower quality fuels. (Slightly heavier blended fuels than this are presently burned in the aircraft derivative engines aboard the EUROLINER class ships). An allowance has been made in the analyses for a fuel washing and treatment plant when using IF60. All maneuvering is done on MGO in all cases.

The economics of the container trade on this route dictate that bunkers are taken for the half-voyage, rather than the round trip. New Orleans list prices are generally lower than those in east coast ports, and these were used for the outboard leg. Inboard bunkers were assumed to be taken in Australia, at costs listed for Sydney. All fuel costs were taken at current (June, 1978) list. An allowance of $0.50/ton was added to the price of IF60 to cover additive and treatment costs. The bunker fuel costs are indicated on Table 13.

TABLE 13

BUNKER FUEL COSTS

	Steam	COGAS		
Type of fuel (for propulsion)	Bunker "C"	MGO	MDO	IF60
Lower heating value, Btu/lb at 100F	17,480 (1)	18,120	17,590	17,508
Specific gravity, average	0.971	0.845	0.880	0.938
Price, $/ton (metric) (2)	79.49	130.50	119.65	90.52(5)
Power output at sea, shp	28,500	28,150	28,150	23,150
Fuel rates at above ratings, lb/shp-hr	0.473	0.315	0.3245	0.326
Days at sea per year	265	265	265	265
Fuel cons. at sea, tons(m)/year	38,900	25,590	26,360	26,480
Days maneuvering per year	9	9	9	9
Fuel cons., maneuvering, tons(m)/year	900	585(3)	585(3)	585(3)
Days stand-by and in port per year	76	76	76	76
Fuel cons., stand-by and in port/year	760	304(4)	304(4)	304(4)
Total fuel cons., tons(m)/year	40,560	26,479	27,249	27,369
Total fuel cost, $1000/year	3,224	3,456	3,270	2,513

(1) Higher heating value: 18,500 lb/hr
(2) Average of Sydney and New Orleans prices
(3) 45 tons (m) of MGO for diesel generators
(4) MGO for diesel generators
(5) Includes $0.50 for additives and treatment cost

While it appears that heating value of residual fuels available on the market approximate the standard, i.e. HHV of 18,500 Btu/lb, information received on the fuel burned by the CALLAGHAN indicate that the LHV of that MDO is approximately 18,400 Btu/lb and that the fuel price in New York was approximately as per the list. If the MDO available on the market has in fact this higher heat content, then the annual fuel cost for the COGAS containership when burning MDO would be $3,131,000.

The COGAS propulsion plant is designed for unattended operation at all times. Engine Department personnel will work a normal eight hour day performing routine tasks, but no watches will be stood. The high level of automation of the gas turbine, the "slaving" of the fully automated steam plant, the use of an unfired heat recovery boiler and of proven components qualify the plant for operation by the smallest crew permitted by regulatory bodies and by union agreements. With an understanding of the maintenance and servicing requirements of the plant, it was determined that an Engine Department of six, as shown in Table 14, is adequate for operating the plant. This is felt to be reasonable and in line with published information on manning scales for other innovative and/or fully automated plants in service.

The base-ship machinery comprises a modern, highly automated, but not unattended steam plant, on which two-man watches are stood. Similar steam container ships actually in service carry a thirteen man Engine Department, but it was felt that a reduction to eleven might be possible. This was taken as a reasonable minimum manning scale as shown in Table 14.

TABLE 14

REASONABLE MINIMUM MANNING SCALES
(ENGINE DEPARTMENT ONLY)

	Steam	COGAS
Chief Engineer	1	1
1st Asst. Engineer	1	1
2nd Asst. Engineer	1	-
3rd Asst. Engineer	2	1
Electrician	1	1
Mechanic	1	1
Oiler	3	-
Wiper	1	1
Total	11	6

The resulting reduction by five men will permit a further reduction in the Steward Department by at least one man. For purposes of this study, this was assumed to be a messman.

Manning costs were determined for each position by a complex but commonly accepted procedure which sums the base wage, supplements for non-watchstanding and for ship category, overtime payments, pension, vacation, and other benefits, for both the man filling the position and his vacation relief. Wage data was taken from labor agreements currently in effect. Subsistence costs were taken directly from data published in 1973 updated tothe present (mid-1978) by the consumer price index. Resulting costs for each position are shown in Table 15.

TABLE 15

ENGINE DEPARTMENT CREW AND SUBSISTENCE COSTS

	Steam	COGAS
Chief Engineer	$135,600	$135,600
1st Asst. Engineer	92,100	92,100
2nd Asst. Engineer	83,300	-
3rd Asst. Engineer A*	71,000	68,400
3rd Asst. Engineer B	71,000	-
Electrician	36,300	36,300
Mechanic	33,900	33,900
Oiler A	30,500	-
Oiler B	30,500	-
Oiler C	30,500	-
Wiper	21,800	21,800
Subtotal	$636,500	$388,100
Less Messman	-	18,200
Plus Subsistence	18,300	8,300
Adjusted Engine Dept. Annual Crew Cost	$654,800	$378,200

* Salary difference due to watch/non-watch status

Insurance costs will largely divide between charges for Hull and Machinery insurance (H&M) and those for Protection and Indemnity insurance (P&I).

H&M rates are frequently quoted as a straight percentage of insured value. It was determined that this rate is not affected by the types of propulsion plant under consideration. The rate is likely to be more seriously affected by other factors influencing or reflecting risk. From published estimates and from source in the marine insurance industry, it was determined that a rate of 1.0% of insured value might produce reasonable H&M annual charges. The insured value of a ship is generally related to its replacement cost. For purposes of this comparison, of new ships, insured value taken to be the capital cost, with or without subsidy as the case may be.

P&I insurance costs are more difficult to model than H&M costs as their dependence on factors related to their ownership and management is greater. P&I costs will be affected by crew size and ship size, but are more related to ship and fleet history, trade, management and ownership. It was determined, however, that P&I costs would not be affected by the contemplated changes in propulsion machinery. From the literature and from sources in the marine insurance industry a method of modelling P&I costs was developed which appears to give realistic values. This was used in estimating the P&I insurance costs shown in Table 16.

Only maintenance costs for propulsion machinery and directly related equipment were compared, there being no reason for hull maintenance requirements to differ between the ships.

Maintenance costs for the steam power plant were estimated using information available from the literature and from data accumulated from previous studies, after updating to current levels.

Maintenance costs for the COGAS power plant were estimated by combining the costs associated with maintenance of the main propulsion gas turbine and those for maintenance of the other components of the power plant. The gas turbine maintenance cost was estimated from information obtained from the manufacturer on the basis of engine rating, type of fuel, and operating profile. These costs represent the total costs to ship operators including parts for routine maintenance and parts and labor for hot section repair and major engine overhauls based upon the estimated engine life for the defined operating conditions. The costs include maneuvering operation and thermal cycles for engine start ups. The maintenance cost for the remainder of the COGAS plant was estimated from information available from the literature and previous studies after allowing for differences in the type of COGAS components, such as heat recovery boiler, generator drive, etc. The latter costs were also compared to those of the steam power plant keeping in mind both the machinery that is common to the two plants and that which is different.

TABLE 16

INSURANCE COST ESTIMATES

	Steam		COGAS	
Estimated ship's cost, $1000	90,000		89,200	
GRT	31,000		31,000	
Complement	36		30	
Construction differential subsidy	None	50%	None	50%
Insured Value, $1000	90,000	45,000	89,200	44,600
H&M annual premium, $1000	900	450	892	446
P&I annual premium, $1000	279	279	261	261
Total annual premium, $1000	1,179	729	1,153	707

The estimates of the maintenance costs (mid-1978) are summarized as follows:

	Steam	COGAS
Annual Maintenance Costs, $1000	127	263*

*Includes an estimated maintenance cost for the gas turbine of $162,000

In addition to the analysis of maintenance costs, a study was conducted of the maintenance requirements of the COGAS plant to determine if they may result in increased downtime compared with that of a steam plant.

Trade patterns used in this study for both the steam ship and the COGAS ship assumed that the ships are in operation 350 days per year, with a total of 15 days off-time each year. The fifteen days are meant to include all off-hire time covering drydocking, scheduled maintenance, as well as breakdowns.

The three principal areas in which COGAS ship maintenance requirements may differ substantially from steamship requirements are: gas turbine, diesel generators and controllable pitch propeller.

Routine checks, tests, filter services, running inspections and adjustments recommended by the manufacturer can be carried out with the engine in service. . A boroscope inspection is recommended at 1200 hour (50 day) intervals, and requires an engine shut-down. A complete boroscope inspection aboard a ship is estimated by the manufacturer to require less than eight hours with two men. These inspections, therfore, can be conducted during the ship stays in port. Hot section components in the LM-2500 have a limited life span and consequently their replacement becomes a maintenance consideration. Hot section repair interval for the LM-2500 at rated output is estimated by the manufacturer to be "in the order of 9000 hours". At the lower output required in the containership application, higher component lives may be anticipated unless they are adversely affected by such other factors as fuel quality. Component lives demonstrated in service over the last eight years aboard the GTS ADM. WILLIAM M. CALLAGHAN support these estimates. When hot section repairs are indicated by boroscope inspection or engine instrumentation, the established procedure is to replace the unit as a module, returning the used module to a depot for overhaul. Manufacturer's estimates, supported by experience aboard the CALLAGHAN and the EUROLINER vessels, point towards elapsed time for removal and replacement on the order of 24 hours. As the exchange need not take place at a shipyard or even at dockside, there is no reason to assume any effect on drydocking cycle of engine repair interval. With proper coordination, engine renewals can be made to coincide with routine port calls to minimize time spent off-hire. There would not appear to be any reason to allot additional time off-hire to the COGAS vessels for gas turbine maintenance.

Although the COGAS vessel relies on diesel generators for ship's service when not underway at sea, running time on each of these units in the trade pattern considered is unlikely to exceed 1200 to 1500 hours per year. Most maintenance requirements are expected to be met by the ship's staff doing day work at sea, supplemented by work during the annual or biennial maintenance period. These units should not cause additional downtime for maintenance.

Controllable pitch propellers have seen increasing application in higher power ranges and larger diameters in the last half dozen years, but total experience with these extends back over a forty year span. A literature search failed to indicate that ships fitted with controllable pitch propellers should be subject to more downtime for routine maintenance or more frequent failures in service than those with fixed pitch propellers.

In view of the foregoing, it would appear that all additional maintenance required for the COGAS plant can be either undertaken during the 350 day operating period, or within the time required during drydocking periods for hull maintenance. Consequently, no additional downtime would appear to be required.

A summary of annual operating costs without and with subsidies, and sensitivity analyses with respect to cost of capital and distillate fuel are shown in Tables 17 and 18 and on Figures 29 and 30.

The total annual operating costs for a C-8 class containership operating between the U.S. East Coast and Australia are estimated to be as shown in Table 17.

TABLE 17

ANNUAL OPERATING COSTS (WITHOUT SUBSIDY)

	Annual Operating Cost ($1000)				
	Steam	COGAS			
Fuel Type	Bunker "C"	MGO	MDO	MDO*	IF60
Charges for Capital Cost	Base	- 68	- 68	- 68	- 58**
Fuel Cost	Base	+232	+ 46	- 93	-711
Crew Cost (Engine Department)	Base	-277	-277	-277	-277
Insurance Cost	Base	- 26	- 26	- 26	- 25**
Maintenance Cost (Machinery)	Base	+136	+136	+136	+217***
Total Cost Difference	Base	- 3	-189	-328	-854

* MDO with LHV of 18,400 Btu/lb

** Capital charges and insurance costs are higher due to the increased cost of the fuel treatment plant

*** The maintenance cost for the gas turbine when burning this light intermediate fuel was increased arbitrarily by 50 percent

The total annual operating costs for a C-8 containership operating between the U.S. East Coast and Australia and subject to construction and operational differential subsidies are estimated to be as indicated in Table 18.

TABLE 18

ANNUAL OPERATING COSTS (WITH SUBSIDIES)

	Subsidy Rate	Annual Operating Cost ($1000)				
		Steam	COGAS			
Fuel Type		Bunker "C"	MGO	MDO	MDO*	IF60
Charges for Capital Cost	50%	Base	- 34	- 34	- 34	- 29**
Fuel Cost	0	Base	+232	+ 46	- 93	-711
Crew Cost (Engine Dept.)	71%	Base	- 80	- 80	- 80	- 80
Insurance Cost	0	Base	- 22	- 22	- 22	- 21**
Maintenance Cost (Machinery)	0	Base	+136	+136	+136	+217***
Total Cost Difference		Base	+232	+ 46	- 93	-624

* MDO with LHV of 18,400 Btu/lb

** Capital charges and insurance costs are higher due to the increased cost of the fuel treatment plant

*** The maintenance cost for the gas turbine when burning this light intermediate fuel was increased arbitrarily by 50 percent

Sensitivity analyses to changes in cost of capital and in cost of distillate fuel (MDO) are shown on Figures 29 and 30. The yearly operating costs were based on a net annual cost of capital of 8.6%. The curves on Figure 29 allow for estimating these costs at other rates. It will be noted that the results are rather insensitive to this parameter.

Similarly the COGAS yearly operating cost differences from the base (steam plant) were based on fuel prices for MGO, MDO, and IF60, see Table 18. The curve on Figure 30 allows for re-evaluating the yearly operating costs assuming changes in the price of MDO fuel oil. Similar curves can be developed to analyze the effect for changes in the price of the other fuel oils. It will be noted that the results are quite sensitive to this parameter.

ECONOMICS AS APPLIED TO A TANKER

An economic analysis was also prepared comparing a COGAS power plant with a steam power plant in a 350,000 DWT tanker. The steam plant is similar to that installed in the containership. The comparison was carried out for a trade route between the Persian Gulf and the Louisiana Offshore Oil Port (LOOP), and resulted in an operating profile of 332 days at 30,000 shp, 9 days discharging and 9 days loading, maneuvering and standing-by. In this case, the capital cost for the installed COGAS plant is $2,200,000 higher than the steam plant mainly because of the large auxiliary boiler required to supply steam to the cargo pump turbines and for cargo heating. The summary of the annual operating costs, assuming construction differential subsidy of 50%, are shown in Table 19.

TABLE 19

ANNUAL OPERATING COSTS (WITH SUBSIDY)

	Annual Operating Cost ($1000)				
	Steam	COGAS			
Fuel Type	Bunker "C"	MGO	MDO	MDO*	IF60
Charges for Capital Cost	Base	+ 95	+ 95	+ 95	+100**
Fuel Cost	Base	+535	+158	- 6	-749
Crew Cost (Engine Department)	Base	-313	-313	-313	-313
Insurance Cost	Base	- 52	- 52	- 52	- 51**
Maintenance Cost (Machinery)	Base	+216	+216	+216	+333***
Total Cost Difference	Base	+481	+104	- 60	-680
Additional Earnings	Base	- 96	- 90	- 94	- 89
Total Difference	Base	+385	+ 14	-154	-769

* MDO with LHV of 18,400 Btu/lb

** Capital charges and insurance costs are higher due to the increased cost of the fuel treatment plant

*** The maintenance cost for the gas turbine when burning this lighter intermediate fuel was increased arbitrarily by 50 percent.

OPERATING COST DIFFERENCES
COGAS VS STEAM

SENSITIVITY TO CHANGES IN
COST OF CAPITAL
CONTAINERSHIP WITH CONSTRUCTION DIFFERENTIAL
AND OPERATING SUBSIDIES

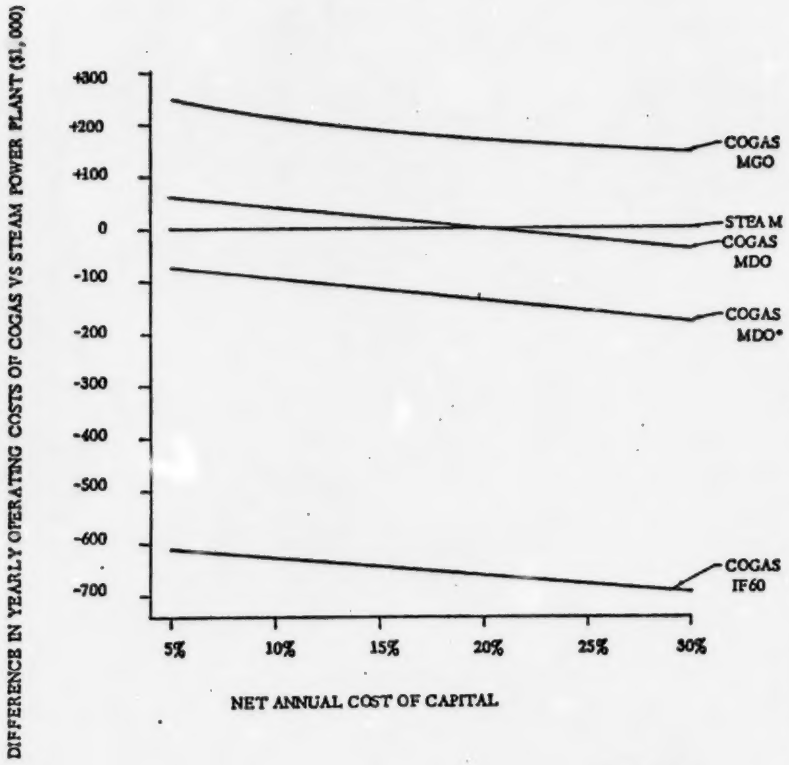


FIGURE 29

OPERATING COST DIFFERENCES
COGAS VS STEAM

SENSITIVITY TO CHANGES IN
COST OF DISTILLATE FUEL
CONTAINERSHIP WITH CONSTRUCTION DIFFERENTIAL
AND OPERATING SUBSIDIES

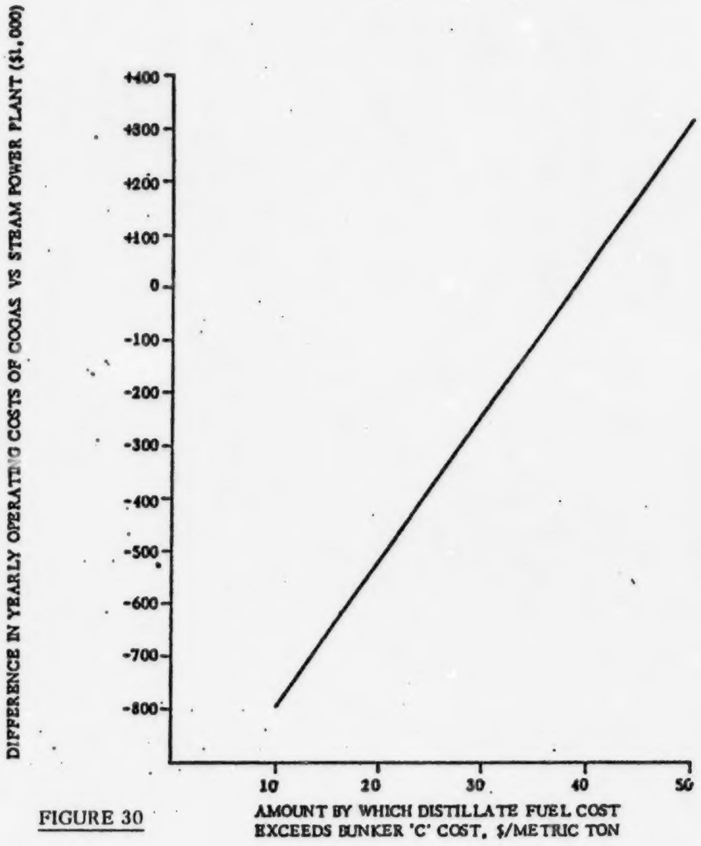


FIGURE 30

ECONOMICS AS APPLIED TO A NAVAL AUXILIARY SHIP

There are several developments in operations of the U.S. Navy that should significantly affect the selection of the type of propulsion plant to be installed on naval auxiliary ships, both those operating with the fleet and those operated point to point by the Military Sealift Command. These developments are:

- The selection of aircraft derivative gas turbine (GE LM-2500) engines for propulsion of the DD-963 and FFG-7 class destroyers and frigates.
- The use of distillate fuel for all ships of the fleet whether steam, diesel or gas turbine propelled.
- Rapid increase in manning cost due to voluntary enlistment in lieu of conscription.
- Rapid increase in crew training cost due to higher personnel turnover rate.
- A continuing decrease in technical competence of personnel.
- A naval policy of endeavoring to overhaul shipboard equipment ashore wherever possible in lieu of aboard ship.
- Increasing cost of propulsion machinery which makes standardization more desirable.

Unlike commercial ships that can select the type of fuel to suit the main propulsion machinery, such as heavy residual fuel for boilers or heavy blended fuel for low-speed cross head type diesel engines, naval auxiliary ships will burn distillate fuel, which is especially suited for aircraft derivative gas turbines, regardless of the type of power plant installed. Thus, in the latter case, the overall economics are very favorable to the installation of COGAS plants because the low fuel rate of these plants is converted into considerable fuel cost savings. A comparison of fuel costs between a COGAS and a steam plant installed on a naval auxiliary ship, such as an AO 177 class oiler, is shown in Table 20.

If a ship of similar type to the AO 177 were operated point to point, as done by the Military Sealift Command, then the savings would be as shown in Table 21.

As described previously, the COGAS plant would utilize the same gas turbine engine now fitted on naval combatant ships. In addition, other components of the COGAS plant, including the steam turbine, heat recovery boiler, condenser, pumps and controls, with the probable exception of the reduction gears, could be standardized and used for propulsion power plants in the power range between 20,000 and 36,700 shp. The commonality of gas turbines, standardization of other components, and simpler propulsion plant offer the following benefits for naval auxiliary ships:

- Reduced crew training, since the crew trained on the combatant ship gas turbine would not require retraining for the gas turbine on the naval auxiliary.
- The COGAS propulsion turbine could be overhauled ashore using the same techniques now adopted for the gas turbines installed on the combatant ships. This arrangement is not possible aboard steam or diesel ships.
- A simplified monitoring and control system facilitating complete automation of the COGAS plant and, therefore, lower engine room manning requirements.
- Lower first cost of COGAS propulsion plants versus steam plants. (Unlike commercial tankers, the auxiliary steam load is relatively low, thus large auxiliary steam boilers are not required.)

TABLE 20

COMPARISON OF YEARLY FUEL COST STEAM VS. COGAS FOR AO 177 CLASS TANKER IN U.S. NAVY PEACETIME OPERATION

Maximum Continuous, SHP	24,000
Assumed hrs. operation at sea (1/3)	2,920
Assumed hrs. standby in port (1/2)	4,380
Assumed hrs. dead ship	1,460
Assumed Percent time at sea	
1/2 Full Speed	35
2/3 Full Speed	35
3/4 Full Speed	15
90% Full Speed	10
Full Speed	5
Assumed price of fuel, $/bbl	18.90
Fuel Consumption, bbls/year, steam	84,250
Fuel Consumption, bbls/year, COGAS	55,900
Savings COGAS, bbls/year	28,350
Fuel Cost Savings COGAS, $/year	535,800
Fuel Cost Savings, 25 year life, 10^6 $	25.6

Life cycle fuel cost savings considers 5% escalation per year in fuel price.

TABLE 21

COMPARISON OF YEARLY FUEL COST STEAM VS. COGAS FOR AO 177 CLASS TANKER IN MILITARY SEALIFT COMMAND OPERATION

Maximum Continuous , SHP	24,000
Assumed 80% year at full power, hrs/yr.	7,008
Assumed 2 weeks dead ship, hrs/yr.	336
Assumed in Port, hrs/yr.	1,416
Assumed price of fuel, $/bbl	18.90
Fuel Consumption, bbls/year, Steam	331,300
Fuel Consumption, bbls/year, COGAS	196,700
Savings COGAS, bbls/year	134,600
Fuel Cost Savings COGAS, 10^6 $/year	2.54
Fuel Cost Savings, 25 year life, 10^6 $	121

Life cycle fuel cost savings considers 5% escalation per year in fuel price.

- The heat recovery boiler can be dry-fired without damage.

- Lower fuel consumption of the COGAS plant at all power ranges compared to a steam plant.

- High reliability of the COGAS plant since it can be operated on the gas turbine alone or the steam plant alone.

- Lower fuel consumption, providing the auxiliary ship with greater cruising range.

- Lower annual fuel cost and crew cost for the COGAS ship versus that presently experienced in the fleet.

CONCLUSIONS

The significant conclusions resulting from this study are:

ENERGY SAVINGS

- The COGAS power plant is significantly more energy efficient, by approximately 30%, when compared to the steam power plant selected for the ships discussed in this paper.

- By inspection it can be determined that the COGAS power plant is more energy-efficient than an open or regenerative cycle gas turbine, any steam plant including reheat steam, or any diesel propulsion system available today.

- If all large U.S. flag ships were fitted with COGAS propulsion systems in lieu of the present-day systems installed, the savings for the ocean-going fleet could amount to approximately 40%.

- The government has expressed a need to conserve energy. If we are to attain this objective, we must install more efficient propulsion power plants such as a COGAS system.

FUEL CONSUMPTION

- The overall fuel consumption (fuel rate) for the COGAS cycle selected at the 30,000 shp rating is about 0.31 lbs/shp-hr when burning No. 2 diesel oil (Marine Gas Oil - MGO).

- The overall fuel consumption of the COGAS power plant fitted with the equipment designed for the 30,000 shp rating is also relatively low at lower power outputs. For example, at 20,000 shp the fuel rate is 0.34 lbs/shp-hr., while at a part load of 10,000 shp the fuel rate is 0.43 lbs/shp-hr.

- When using water injection (water mixed with fuel), the COGAS plant produces about 36,500 shp and the fuel rate is only slightly higher than at 30,000 shp, that is, 0.32 lbs/shp-hr.

FUEL TYPES

- The COGAS power plant, since it employs an aircraft derivative engine, is capable of burning only (MGO), clean marine diesel oil (MDO), or other distillate fuels.

- Testing is now underway exploring the possibility of burning blended fuel oil (IF 60). This fuel would be treated to inhibit and/or remove contaminants. Blended treated fuel oil is already being burned in aircraft derivative engines aboard the "Euroliner" class ships.

FUEL COSTS

- For the containership and tanker included in the economic study, the yearly cost of fuel for the steam plant when burning Bunker C fuel is less than the COGAS plant when burning MGO, while on the other hand when the COGAS plant is burning MDO, the yearly fuel cost is somewhat less than for the steam plant. However, significant savings in the range of 20% can be made of a blended fuel such as IF 60 is burned.

- The price spread between MGO or MDO and Bunker C has been widening in the last two years which has made the COGAS plant less attractive from a fuel cost standpoint. (For example, in the last two years the differential between MGO/MDO and Bunker C has increased by about 15%).

- A large portion of the crude oil processed today is performed by small unsophisticated refineries (type 2 plants), with the result that over half of each barrel of crude is turned into residual fuel. This large supply of residual has a depressing effect on the price as compared with distillates. Unless refining practices are upgraded, it is likely that the current differential price between residual and distillate will remain or possibly could be greater.

PACKAGE POWER PLANT CONCEPT

- The COGAS power plant lends itself well to a wide range of propulsion powers while using all the same components (20,000 to 30,000 shp), thus allowing significant standardization.

- The fewer COGAS power plant components required and simpler piping systems should further provide for standardization and low cost of installation.

- The power range of the COGAS plant can be extended to 36,700 shp by the use of water injection while still using the same basic power plant components.

SPACE AND WEIGHT

- The space requirements for the COGAS plant are the same or marginally less than that required for a steam plant.

- The weight of the COGAS plant is significantly less than that of a steam plant.

PROPULSION CONTROL SYSTEM

- The COGAS control system lends itself to single lever bridge control and unattended engine room operation during steady state, maneuvering and standby periods.

- Use can be made of proven control systems and concepts to provide a highly reliable and low cost system.

- The control system maximizes the heat recovery from the gas turbine exhaust by "slaving" the steam plant to the gas turbine.

- The control system envisions the use of a reversible controllable pitch propeller, thus utilizing a proven concept.

- The COGAS plant has two complete independent power systems, that is, the power plant can be operated by the gas turbine alone or by the steam plant alone.

- The heat recovery boiler is inherently a more reliable component than an oil-fired steam boiler since dry firing of the unit will not be detrimental to the equipment. Further, the steam pressures and temperatures utilized in the system are lower than conventional steam equipment used today, thus improving reliability.

- Fewer components are required in the COGAS plant than in a conventional steam plant, thus improving reliability.

- The LM-2500 gas turbine has inherent high reliability as has been proven by in-service operation on the CALLAGHAN and aboard the new U.S. naval combatant ships.

ECONOMIC COMPARISON RE A CONTAINERSHIP AND A TANKER

- Capital Cost for a COGAS plant is less than for a comparable steam plant when applied to a containership. However, the reverse is true when considering a tanker due to the need for a large auxiliary boiler for pumping cargo and tank heating.

- Crew and Subsistence Costs are less for both the containership and tanker since the simpler COGAS power plant permits a reduction in engine room crew, as well as a corresponding small reduction in the Stewards' Department.

- Insurance Costs are somewhat less for the containership employing a COGAS plant but the reverse is true for the tanker since insurance costs are related to the cost of the ship.

- Maintenance Costs are higher for the COGAS power plant as it relates to both the containership and tanker. This is due to the higher maintenance cost associated with the gas turbine.

- Annual Operating Costs are lower for the COGAS plant as applied to the containership and tanker when burning clean marine diesel oil and are substantially lower when burning a blended fuel. When burning marine gas oil, the reverse is true.

U.S. NAVAL AUXILIARY SHIPS

- The COGAS plant installed on naval auxiliaries offers significant savings in first cost, fuel cost, crew cost and crew training requirements.

- The standardized COGAS propulsion plant can be fitted in most all large size auxiliary ships since the standardized design covers a power range from 20,000 to 36,700 shp.

ACKNOWLEDGMENT

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BIBLIOGRAPHY

A list of technical studies and papers on the subject of combined gas and steam turbine plants is included in Appendix C.

APPENDIX A

ECONOMIC EVALUATION OF COGAS CYCLES

The economic evaluation of the various COGAS cycles was conducted to establish the most economic COGAS arrangement. Variations in throttle steam conditions, condenser exhaust pressures and boiler pinch point temperatures were considered. The resultant changes in capital costs and fuel rates were referred to a "base" cycle described below.

For most all of the cycles considered, the equipment is essentially the same with only minor effects on maintenance and repair costs, manning, insurance, etc. Consequently, the economic evaluation narrows down to a comparison on the basis of fuel costs and capital costs only. Capital cost increases justified by a one percent decrease in fuel consumption are detailed in Table A-1.

The simplest cycle considered is the single pressure boiler cycle, with throttle conditions of 450 psig/850F, condenser pressure of 2 in. Hg abs, and a 30F pinch point. This cycle was chosen as the base cycle for this analysis. All changes from this base cycle were found to require greater capital cost but not all were feasible, and not all resulted in improvements in the fuel rate. Specifically, almost all of the 850 psig single pressure cycles resulted in poorer fuel rates (see Table A-2, runs 13 to 18) and were eliminated from further consideration for that reason, while 850 psig multi-pressure boiler cycles were found to be technically unfeasible with the constraints considered because of insufficient pinch point at the intermediate steam generating bank (see Table A-3, runs 13 to 18 and run 21).

To evaluate the remaining cycles it was necessary to select parameters for comparing changes in fuel cost with changes in initial capital cost. In view of the probability of continually increasing rises in the price of petroleum fuels it was determined that the comparison must be made by returning life cycle fuel costs to their present worth. Present worths of annual savings per percent improvement in fuel rate for 260 and 320 days per year of full power operation (reflecting a containership and a tanker respectively) for the particular fuel required (No. 2-D, Marine Gas Oil) and for a 20 year life of the plant were developed for 5% and 10% annually compounded increases in fuel costs and cost of capital of 15% and 30% per annum. Fuel costs have been escalated at the rate indicated from the present time to the commencement of service and on through the life of the ship. The resulting sums are shown on Table A-1 as capital cost increase justified by a one percent decrease in fuel consumption. These sums are hereinafter referred to as the "Justifiable Cost Parameter", or JCP.

Capital cost estimates for the principal plant components most affected by variations in the heat cycle are shown in Tables A-2 and A-3 as increments from the base cycle (Run 4 on Table A-2) cost. These estimates were derived from cost information obtained from various equipment manufacturers for varying sets of operating conditions (throttle pressure and temperature, condenser pressure, pinch point, flows, etc.) and for both single and mulit-pressure cycles. In the case of multi-pressure cycle, an estimated cost for piping systems was added to allow for the increased amount of interconnecting piping, valves and controls associated with the additional steam drum, steam generating bank, and the steam turbine injection line.

The procedure adopted to compare the merit of the increased capital cost to decreases in annual operating cost was to take the ratio of the total capital cost increment for each cycle to the percent improvement in fuel rate. These ratios are shown as the last column in Tables A-2 and A-3, the change in capital cost per one percent decrease in fuel consumption, hereinafter referred to as the Cost Parameter, or CP. These values may be compared to the appropriate Justified Cost Parameter (see Table A-1) to determine if the investment is economically justified for any particular combination of fuel costs, capital costs and days at sea. In comparison with each other, these values indicate the cost effectiveness of capital cost dollars, the lower values indicating highest cost effectiveness.

A comparison of the single pressure cycles with each other on this basis and on the basis of fuel rate indicates that the most significant improvements in the cycle result from increasing the superheat temperature and lowering the pinch point. Lowering back pressure will improve the fuel rate but only with large capital cost increases. Raising the throttle pressure proved counter-productive.

The cycle identified as Run 23 on Tables A-2 and A-4 (450 psig/885F at the throttle, 2 in. Hg., abs. backpressure, 20F pinch point) was selected as the optimum single pressure cycle, yielding a slight gain in fuel rate for a modest investment in capital cost. Additionally, a very attractive feature of the single pressure cycle is the simplicity of the design.

There is an economic incentive to develop the more complex mulit-pressure cycle. However, it is felt that the simple single pressure cycle would be more attractive to prospective shipowners who are willing to install a COGAS plant for the first time. Also, if it is found that the gas turbine can burn less expensive fuels than the MGO used in the evaluation, the economic incentive to install the more complex multi-pressure plant would be reduced considerably.

The weight and space associated with the various cycles have not been considered in this economic evaluation since it is believed that they will not be appreciable enough to affect the results.

A comparison of the various cycles considered has been plotted as related to the steam turbine throttle pressure and overall ship's fuel rate for both the single pressure and mulit-pressure heat recovery boiler cycles. These are shown on Figures A-1 and A-2. These curves clearly show that for the single pressure boiler throttle pressures in the range of 400-500 psig are the most economical, while for the multi-pressure boiler throttle pressures in the range of 700 and 800 psig are more desirable. The curves also show that higher superheat temperature, improved vacuum and/or lowering of the pinch points increases economy as well.

TABLE A-1

CAPITAL COST INCREASE JUSTIFIED BY A ONE PERCENT DECREASE IN FUEL CONSUMPTION

Type of fuel . . . . Marine Gas Oil
Present Cost of fuel,(end '77, N. Y. Harbor) $17.44/bbl
Life of Ship . . . . 20 years
Commencement of service 1981

Cost of Capital to Shipowner %	Annual Rise in Fuel Price % of Previous Year*	Operating Profile Days of Full Power Operation Per Year	Capital Cost Increase justified by 1% Decrease of Fuel Consumption "ICP", ($1000)
15	0	260	220
30	0	260	116
15	0	320	270
30	0	320	143
15	5	260	356
30	5	260	168
15	5	320	439
30	5	320	207
15	10	260	604
30	10	260	247
15	10	320	743
30	10	320	304

* Fuel costs have been escalated at the rate indicated from the present time to the commencement of service and on through the life of the ship.

TABLE A-2

SINGLE PRESSURE CYCLE

CHANGE IN CAPITAL COSTS AND FUEL RATE DUE TO VARIATION IN CYCLE CONDITIONS

Run	Throttle Psig	F	Exh in Hg	$\Delta T_1/\Delta T_2$ F	Cost Increment from Base, ($1000) Blr	Turb	Cond	Circ	AirEj	Total	Corrected Fuel Rate	%Impr. in Fuel Rate	$1000 % F R
1	450	850	1.5	40/30	4	17	63	32	4	120	.3066	.20	600
2	450	885	1.5	40/30	8	17	61	32	4	122	.3058	.45	271
3	450	920	1.5	40/30	19	31	59	32	4	145	.3052	.65	223
4	450	850	2.0	40/30	-	-	-	-	-	Base	.3072	Base	Base
5	450	885	2.0	40/30	4	-	-1	-	-	3	.3065	.23	13
6	450	920	2.0	40/30	15	12	-2	-	-	25	.3059	.42	60
7	600	850	1.5	35/30	12	12	57	32	4	117	.3071	.03	3900
8	600	885	1.5	35/30	16	12	55	32	4	119	.3066	.20	595
9	600	920	1.5	35/30	27	29	54	32	4	146	.3060	.39	374
10	600	850	2.0	35/30	-	-	-	-	-	-	.3079	Neg.	-
11	600	885	2.0	35/30	-	-	-	-	-	-	.3073	"	-
12	600	920	2.0	35/30	23	12	-6	-	-	29	.3067	.16	181
13	850	850	1.5	30/30	-	-	-	-	-	-	.3092	Neg.	-
14	850	885	1.5	30/30	-	-	-	-	-	-	.3088	"	-
15	850	920	1.5	30/30	-	-	-	-	-	-	.3083	"	-
16	850	850	2.0	30/30	-	-	-	-	-	-	.3101	"	-
17	850	885	2.0	30/30	-	-	-	-	-	-	.3097	"	-
18	850	920	2.0	30/30	-	-	-	-	-	-	.3092	"	-
19	450	885	1.5	40/20	32	17	65	32	4	150	.3044	.91	165
20	600	885	1.5	35/20	40	12	59	32	4	147	.3050	.72	204
21	850	885	1.5	30/20	50	14	54	32	4	154	.3070	.07	2200
22	600	885	1.5	30/30	40	12	56	32	4	144	.3062	.33	436
23	450	885	2.0	40/20	28	-	-	-	-	28	.3050	.72	39

TABLE A-3

MULTI-PRESSURE CYCLE

CHANGE IN CAPITAL COSTS AND FUEL RATE DUE TO VARIATION IN CYCLE CONDITIONS

Run	Throttle Psig	Throttle F	Exh in Hg	$\Delta T_1/\Delta T_2/\Delta T_3$ F	Cost Increment from Base. ($1000) Blr	Turb	Cond	Circ	AirEj	Pip.Sys	Total	Corr'd Fuel Rate	% Impr. in Fuel Rate	$1000 % F R
1	450	850	1.5	40/30/50	84	27	69	32	4	44	260	.3025	1.53	170
2	450	885	1.5	40/30/50	93	27	68	32	4	44	268	.3010	2.02	133
3	450	920	1.5	40/30/50	117	41	68	32	4	44	306	.3001	2.31	132
4	450	850	2.0	40/30/50	77	10	4	-	-	44	135	.3027	1.46	92
5	450	885	2.0	40/30/50	84	10	4	-	-	44	142	.3014	1.88	76
6	450	920	2.0	40/30/50	109	22	3	-	-	44	178	.3002	2.27	78
7	600	850	1.5	35/30/50	204	22	68	32	4	44	371	.3009	2.05	182
8	600	885	1.5	35/30/50	215	22	66	32	4	44	383	.2996	2.47	155
9	600	920	1.5	35/30/50	237	39	65	32	4	44	421	.2983	2.90	145
10	600	850	2.0	35/30/50	197	8	4	-	-	44	253	.3011	1.98	123
11	600	885	2.0	35/30/50	204	8	3	-	-	44	259	.2998	2.40	108
12	600	920	2.0	35/30/50	227	22	2	-	-	44	295	.2987	2.76	107
13	850	850	1.5	30/30/50										
14	850	885	1.5	30/30/50	Not Feasible Due To									
15	850	920	1.5	30/30/50	Insufficient Pinch									
16	850	850	2.0	30/30/50	Point at the Intermediate									
17	850	885	2.0	30/30/50	Steam Generating Bank									
18	850	920	2.0	30/30/50										
19	450	883	1.5	40/20/50	117	27	69	32	4	44	293	.3019	1.73	169
20	600	885	1.5	35/20/50	239	22	67	32	4	44	408	.3003	2.25	181
21	850	885	1.5	30/20/50	Not Feasible									
22	600	885	1.5	30/30/50	220	22	66	32	4	44	388	.2997	2.44	159

TABLE A-4

SINGLE PRESSURE CYCLE - RUN 23

```
HEAT BALANCE OUTPUT WITH CALCULATED STACK TEMP. - NO BLEEDING
-------------------------------------------------------------

F1     = 58787.96 FTH   = 58647.96 FTG   =     0.00 FFP   =     0.00
FL     =   340.00 FEXST= 58420.48 GLAND=   227.47 FTGLN=     0.00
GL+VT=   327.47 FMU   =   340.00 FEVAP=   543.43 FDUMP=     6.32
FMCND= 59637.71 VENT  =   100.00 F2    =  9001.95 F1+F2= 67789.92
BLEED=     0.00

PLFT  =    25.00 PRES1=    25.00 PRES2=   491.00 PSH   =   477.00
PTH   =   465.00 PMCND=     2.06

TSH   =   890.00 TTH   =   885.00 TDFTL=   240.06 TEV   =   465.14
TEC   =   425.14 TDECO=    40.00 TPR1L=   240.06 PINPT=    20.00
T1    =   977.84 T2    =   485.14 TSTK  =   317.95

CP1   =   0.2857 CP2   =   0.2692 CPSTK=   0.2620

HSH   =  1461.42 HTH   =  1459.17 SLEP  =  1046.74 HGLND=  1234.20
HTGLN=     0.00 HMCND=    69.10 HGLCF=    75.06 HDFTG=  1160.54
HDFTL=   208.41 HPR1L=   208.41 HPR1G=  1160.54 HFPX  =   210.43
HMFPX=     0.00 HTECO=   402.43 HPR2L=   447.26 HPR2G=  1404.44

STEAM TURBINE CONDITIONS
-------------------------------------------------------------

SHPR  =  7500.00 SPDR  =   100.00 SHPC  =  9158.56 SPDD  =   106.88
STMRT=   6.4036 FLOWC= 58648.06

ATTACHED GENERATOR CONDITIONS
-------------------------------------------------------------

KWBAS=   365.25 BCPKW=     0.41 CPKW  =     3.60 VPKW  =    15.00
FPKW  =    47.57 TOTKW=   431.84 HPKW  =   622.68

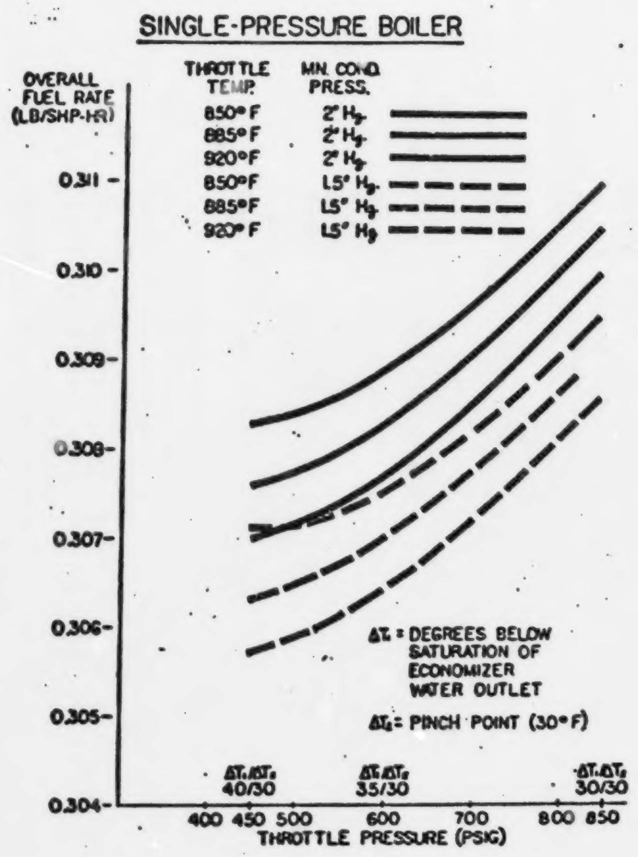
GAS TURBINE CONDITIONS
-------------------------------------------------------------

SHP   = 22167.60 SPEED=     3600 TOUT  =    75.00 XIN   =     3.00
XOUT  =    13.00 HUMGR=   117.00 LHV   = 18120.00 T1    =   977.84
CP1   =   0.2857 SPFCS=   0.4148 WGAS  =466217.06

OVERALL FUEL RATE = 0.3056
--------------------------

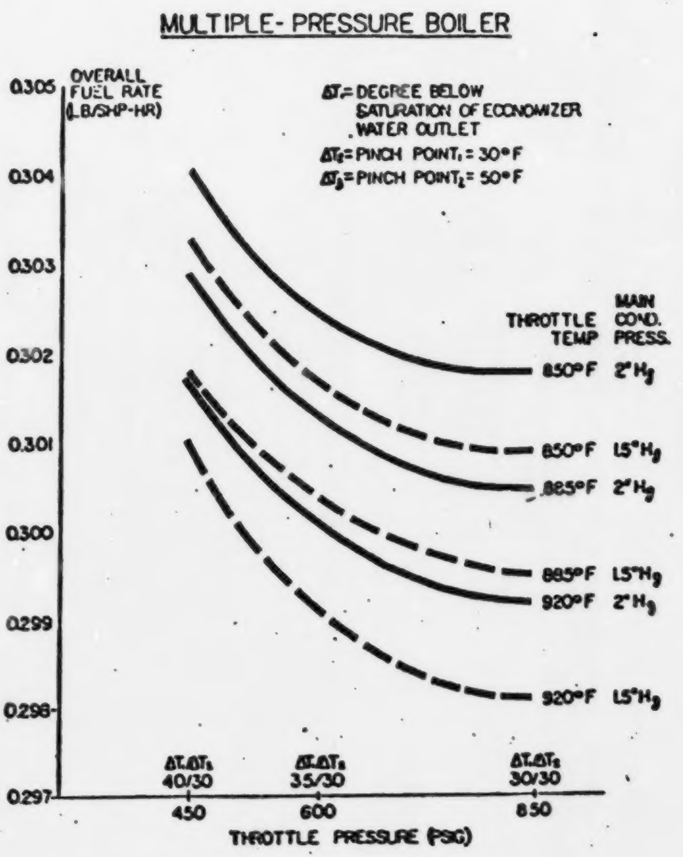
TOTAL SHP = 30703.48
--------------------
```

NOTE: For definition of the above symbols, see List of Abbreviations in Appendix B.



OVERALL FUEL RATE VS. THROTTLE PRESSURE

FIGURE A-1



OVERALL FUEL RATE VS. THROTTLE PRESSURE

FIGURE A-2

APPENDIX B

SELECTED COGAS CYCLE PERFORMANCE

Included for reference on the following sheets is the estimated performance for the selected cycle at various levels of output. Also included is a list of abbreviations so that the various temperatures, pressures, flows, powers and other ratings can be identified.

LIST OF ABBREVIATIONS FOR HEAT BALANCE COMPUTER OUTPUT SHEETS

Flows-LB/HR

F1	The superheater flow generated by the heat recovery boiler
FTH	The throttle flow to the single casing steam turbine
FTG	Not Applicable (steam flow to turbogenerator)
FFP	Not Applicable (steam flow to feed pump turbine)
FL	The system fluid losses equal to 0.5% of the sum of F1 and F2
FEXST	The exhaust flow from the steam turbine to the main condenser
GLAND	The total gland leakoff steam = 200 + 0.3% SHPD
FTGLN	Not Applicable (turbogenerator gland leakoff)
GL & VT	The gland and vent flow combined
FMU	The make-up flow to the main condenser, equal to the system fluid losses
FEVAP	The required steam flow to the distilling plant considered to be a two-stage flash type distilling plant
FDUMP	The excess steam dumped to the main condenser from the D.F.T.
FMCND	The condensate flow from the main condenser
VENT	The saturated vent steam from the D.F.T. to the Gland Leakoff condenser
F2	The flow from the D.F.T. to the stack heater returning to the D.F.T.
F1 & F2	The superheater and stack heater flow combined
BLEED	The bleed steam required to maintain the D.F.T. at its design conditions
GPD	Gallons per Day-Fresh water from the distilling plant
WWPPH	Weight of water pounds per hour injection to gas turbine

Pressures-PSIA

PDFT	D.F.T. pressure
PRES1	Not Applicable (same as PDFT)
PRES2	Boiler drum pressure
PSH	Superheater outlet pressure
PTH	Throttle pressure
PMCND	Main condenser vacuum (in Hg)

Temperatures - °F

TSH	Superheater outlet temperature
TTH	Throttle temperature
TDFTL	D.F.T. temperature
TEV	The temperature of the steam flow leaving Boiler (Drum Sat. Temp.)
TEC	The temperature of the feed flow leaving the economizer
TDECO	The degrees below saturation of the economizer feed flow outlet
TBLED	Bleed temperature if applicable
PINPT	Temperature difference between gas temperature (T2) and Boiler drum saturation temperature (TEV)
T2	Exhaust gas temperature at a point between the steam generating bank and the economizer
TSTK	Exhaust gas temperature after the stack heater
T1A	Exhaust gas temperature between the superheater and the steam generating bank
T2A	Exhaust gas temperature between the economizer and the stack heater

Specific Heat-BTU/LB-°F

CP2	Specific heat of exhaust gas corresponding to T2
CPSTK	Specific heat of exhaust gas corresponding to TSTK
CP1A	Specific heat of exhaust gas corresponding to T1A
CP2A	Specific heat of exhaust gas corresponding to T2A

Enthalpies-BTU/LB

HSH	Superheater outlet
HTH	Throttle
SLEP	Calculated endpoint enthalpy to the main condenser
HGLND	Calculated enthalpy of the GLAND flow
HTGLN	Not applicable (turbogenerator gland leakoff)
HMCND	Main condenser condensate flow enthalpy
HGLCF	The calculated enthalpy of the condensate flow leaving the gland leakoff condenser
HDFTG	D.F.T. saturated vapor enthalpy
HDFTL	D.F.T. saturated liquid enthalpy
HPR1L	Not applicable (same as D.F.T.)
HPR1G	Not applicable (same as D.F.T.)
HFPX	Enthalpy of the flow leaving the main feed pump as a compressed liquid.
HBLED	Bleed enthalpy if applicable
HTECO	Enthalpy of the feed flow leaving the economizer
HPR2L	Saturated liquid enthalpy at the Boiler drum pressure
HPR2G	Saturated vapor enthalpy at the Boiler drum pressure

Under the heading "Steam Turbine Conditions", the following definitions apply:

SHPR	Rated steam turbine shaft horsepower
SPDR	Steam turbine speed at rated shaft horsepower - RPM
SHPD	Design steam turbine shaft horsepower
SPDD	Steam turbine speed at design shaft horsepower - RPM
STMRT	Calculated non-extraction steam rate at design shaft horsepower
FLOWC	This is a heat balance check to ensure that the throttle flow is equal to the actual flow used in calculating the non-extraction steam rate and the endpoint
TEXL	Total Exhaust Loss HP

Under the heading "Attached Generator Conditions", the following definitions apply:

TOTAL KW LOAD	The total KW load to be produced by the attached generator during full power operation
GENERATOR HORSEPOWER	The horsepower requirement for the KW load

Under the heading "Gas Turbine Conditions", the following definitions apply:

SHP	Total corrected shaft horsepower produced by the gas turbine
SPEED	Power turbine speed
TOUT	Outside ambient air temperature °F
XIN	Inlet losses inches of H_2O
XOUT	Outlet losses inches of H_2O
HUMGR	Grains of humidity based on 85% relative humidity
LHV	Lower heating value of the fuel oil BTU/LB
T1	Exhaust gas temperature °F leaving the gas turbine
CP1	Specific heat of exhaust gas corresponding to T1
SPFCS	Specific fuel consumption LBS/BHP-HR
WGAS	Gas flow from the gas turbine LB/HR

OVERALL FUEL RATE	The COGAS plant fuel rate LBS/SHP-HR
TOTAL SHP	The total COGAS plant shaft horsepower (SHPD - Generator Horsepower + SHP)

Additional abbreviations for previous formats of the heat balance output sheets.

TPR1L	Not applicable (temperature same as D.F.T.)
HMFPX	Not applicable (enthalpy of feed pump turbine exhaust)
SPDR	Steam turbine rated speed (percent of rated speed)
SPDD	Steam turbine design speed (percent of rated speed)

Under the heading "Attached Generator Conditions", there are the Generator load requirements for the cycle based on applying the formula in SNAME T&R Bulletin 3-28, Gas Turbine Power Plant Performance Practices, modified to include the electrically driven steam cycle equipment. The definitions follow.

KWGAS	KW load from the formula
BCPKW	KW load for the Stack Heater circulation pump
CPKW	KW load for the main condensate pump for FMCND
VPKW	KW load for the main vacuum pump
FPKW	KW load for the main feed pump
TOTKW	Total KW load
HPKW	Corresponding shaft horsepower requirement for TOTKW

```
CASE 1 - 100 PC POWER - 11/14/78 - J.M. - 5615 - 800 KW LOAD
------------------------------------------------------------
HEAT BALANCE OUTPUT WITH INPUT STACK TEMP. - DUMPING
------------------------------------------------------------

F1    = 56521.44 FTH   = 56195.44 FTG   =     0.00 FFP   =     0.00
FL    =   326.00 FEXST= 55968.56 GLAND=   226.88 FTGLN=     0.00
GL+VT=   326.38 FMU   =   326.00 FEVAP=   534.02 FDUMP=   131.90
FFCLD= 57267.36 VENT  =   100.00 F2    =  8775.84 F1+F2= 65297.28
BLEED=     0.00 GPD   =  2269.75 WWPPH=     0.00

PDFT =     25.00 PRES1=    25.00 PRES2=   491.00 PSH   =   477.00
PTH  =    465.00 PMCND=     2.00

TSH  =    690.00 TTH   =   885.00 TDFTL=   240.06 TEV   =   465.14
TEC  =    439.14 TDECO=    26.00 TBLED=     0.00 PINPT=    18.00
T2   =    483.14 TSTK  =   311.00 T1A   =   849.68 T2A   =   383.50

CP2  =    0.2577 CPSTK=   0.2500 CP1A  =   0.2685 CP2A  =   0.2544

HSH  =   1461.42 HTH   =  1459.17 SLEP  =  1037.41 HGLND=  1290.47
HTGLN=      0.00 HMCND=    69.10 HGLCF=    75.28 HDFTG=  1160.54
HDFTL=    208.41 HPR1L=   208.41 HPR1G=  1160.54 HFPX  =   210.43
HBLED=      0.00 HTECO=   417.81 HPR2L=   447.26 HPR2G=  1204.44

STEAM TURBINE CONDITIONS
------------------------------------------------------------

SHPR =  9000.00 SPDR  =  5500.00 SHPD  =  8961.55 SPDU  =  5500.00
STMRT=   6.2713 FLOWC= 56193.85 TEXL  =   332.26

ATTACHED GENERATOR CONDITIONS
------------------------------------------------------------

TOTAL KW LOAD =   800.00
GENERATOR HORSEPOWER =  1184.41

GAS TURBINE CONDITIONS
------------------------------------------------------------

SHP  = 22167.60 SPEED=     3600 TOUT  =    75.00 XIN   =     3.00
XOUT =    12.80 HUMGR=   117.00 LHV   = 18120.00 T1    =   966.30
CP1  =   0.2738 SPFC8=   0.4116 WGAS  =468719.93

OVERALL FUEL RATE = 0.3111
-------------------------

TOTAL SHP = 29944.74
--------------------
```

NOTE: For definition of the above symbols, see list of abbreviations

```
CASE 2 - 90 PC POWER - 11/14/78 - J.M. - 5615 - 500 KW LOAD
------------------------------------------------------------
HEAT BALANCE OUTPUT WITH INPUT STACK TEMP. - DUMPING
------------------------------------------------------------

F1    = 52789.66 FTH   = 52483.66 FTG   =     0.00 FFP   =     0.00
FL    =   306.00 FEXST= 52256.75 GLAND=   224.89 FTGLN=     0.00
GL+VT=   324.89 FMU   =   306.00 FEVAP=   520.58 FDUMP=   201.33
FFCFG= 53611.57 VENT  =   100.00 F2    =  8363.50 F1+F2= 61153.17
BLEED=     0.00 GPD   =  2231.63 WWPPH=     0.00

PDFT =     25.00 PRES1=    25.00 PRES2=   457.00 PSH   =   445.00
PTH  =    435.00 PMCND=     1.93

TSH  =    882.00 TTH   =   877.00 TDFTL=   240.06 TEV   =   457.83
TEC  =    433.83 TDECO=    24.00 TBLED=     0.00 PINPT=    16.00
T2   =    473.83 TSTK  =   307.00 T1A   =   834.59 T2A   =   379.20

CP2  =    0.2574 CPSTK=   0.2496 CP1A  =   0.2677 CP2A  =   0.2542

HSH  =   1458.28 HTH   =  1455.98 SLEP  =  1037.00 HGLND=  1288.38
HTGLN=      0.00 HMCND=    67.92 HGLCF=    74.47 HDFTG=  1160.54
HDFTL=    208.41 HPR1L=   208.41 HPR1G=  1160.54 HFPX  =   210.28
HBLED=      0.00 HTECO=   411.96 HPR2L=   438.92 HPR2G=  1204.57

STEAM TURBINE CONDITIONS
------------------------------------------------------------

SHPR =  9000.00 SPDR  =  5500.00 SHPD  =  8296.54 SPDD  =  5500.00
STMRT=   6.3236 FLOWC= 52484.31 TEXL  =   324.96

ATTACHED GENERATOR CONDITIONS
------------------------------------------------------------

TOTAL KW LOAD =   800.00
GENERATOR HORSEPOWER =  1184.41

GAS TURBINE CONDITIONS
------------------------------------------------------------

SHP  = 19950.83 SPEED=     3600 TOUT  =    75.00 XIN   =     2.70
XOUT =    11.70 HUMGR=   117.00 LHV   = 18120.00 T1    =   947.30
CP1  =   0.2729 SPFC8=   0.4225 WGAS  =448920.00

OVERALL FUEL RATE = 0.3178
-------------------------

TOTAL SHP = 27062.96
--------------------
```

```
CASE 3 - 80 PC POWER - 11/14/78 - J.M. - 5615 - 800 KW LOAD
-----------------------------------------------------------
HEAT BALANCE OUTPUT WITH INPUT STACK TEMP. - DUMPING
-----------------------------------------------------------

F1     = 49390.92 FTH   = 49104.92 FTG   =     0.00 FFP   =     0.00
FL     =   286.00 FEXST= 48881.89 GLAND=   223.03 FTGLN=     0.00
GL+VT=   323.03 FMU   =   286.00 FEVAP=   507.14 FDUMP=   114.34
FPCND= 50112.40 VENT  =   100.00 F2    =  7832.64 F1+F2= 57223.57
BLEED=     0.00 GPD   =  2174.00 WWPPH=     0.00

PDFT  =    25.00 PRES1=    25.00 PRES2=   427.00 PSH   =   416.00
PTH   =   407.70 PMCND=     1.84

TSH   =   872.00 TTH   =   867.00 TDFTL=   240.06 TEV   =   451.02
TEC   =   430.02 TDECO=    21.00 TBLED=     0.00 PINPT=    14.00
T2    =   465.02 TSTK  =   303.00 T1A   =   821.39 T2A   =   373.99

CP2   =   0.2570 CPSTK=   0.2491 CP1A  =   0.2670 CP2A  =   0.2538

HSH   =  1454.01 HTH   =  1451.66 SLEP  =  1036.47 HGLND=  1265.58
HTGLN=     0.00 HMCND=    66.34 HGLCF=    73.29 HDFTG=  1160.54
HDFTL=   208.41 HPR1L=   208.41 HPR1G=  1160.54 HFPX  =   210.14
HBLED=     0.00 HTECO=   407.78 HPR2L=   431.17 HPR2G=  1204.56

STEAM TURBINE CONDITIONS
-----------------------------------------------------------

SHPR  =  9000.00 SPDR  =  5500.00 SHPD  =  7674.04 SPDD  =  5300.00
STMRT=   6.3943 FLOWC= 49103.35 TEXL  =   318.19

ATTACHED GENERATOR CONDITIONS
-----------------------------------------------------------

TOTAL KW LOAD =   800.00
GENERATOR HORSEPOWER =  1184.41

GAS TURBINE CONDITIONS
-----------------------------------------------------------

SHP   = 17734.07 SPEED=      3600 TOUT  =    75.00 XIN   =     2.50
XOUT  =    10.60 HUMGR=   117.00 LHV   = 18120.00 T1    =   930.30
CP1   =   0.2720 SPFCS=   0.4366 WGAS  =428400.06

OVERALL FUEL RATE = 0.3262
--------------------------

TOTAL SHP = 24223.70
--------------------
```

```
CASE 4 - 70 PC POWER - 11/14/78 - J.M. - 5615 - 800 KW LOAD
-----------------------------------------------------------
HEAT BALANCE OUTPUT WITH INPUT STACK TEMP. - DUMPING
-----------------------------------------------------------

F1     = 45935.17 FTH   = 45668.17 FTG   =     0.00 FFP   =     0.00
FL     =   267.00 FEXST= 45447.02 GLAND=   221.14 FTGLN=     0.00
GL+VT=   321.14 FMU   =   267.00 FEVAP=   494.37 FDUMP=   152.26
FPCND= 46681.80 VENT  =   100.00 F2    =  7420.15 F1+F2= 53355.33
BLEED=     0.00 GPD   =  2119.26 WWPPH=     0.00

PDFT  =    25.00 PRES1=    25.00 PRES2=   395.00 PSH   =   388.00
PTH   =   381.20 PMCND=     1.76

TSH   =   863.00 TTH   =   858.00 TDFTL=   240.06 TEV   =   444.09
TEC   =   425.09 TDECO=    19.00 TBLED=     0.00 PINPT=    13.00
T2    =   457.09 TSTK  =   299.00 T1A   =   809.69 T2A   =   369.96

CP2   =   0.2567 CPSTK=   0.2487 CP1A  =   0.2664 CP2A  =   0.2534

HSH   =  1450.27 HTH   =  1447.88 SLEP  =  1037.07 HGLND=  1263.55
HTGLN=     0.00 HMCND=    64.88 HGLCF=    72.29 HDFTG=  1160.54
HDFTL=   208.41 HPR1L=   208.41 HPR1G=  1160.54 HFPX  =   213.01
HBLED=     0.00 HTECO=   402.36 HPR2L=   423.47 HPR2G=  1204.43

STEAM TURBINE CONDITIONS
-----------------------------------------------------------

SHPR  =  9000.00 SPDR  =  5500.00 SHPD  =  7041.60 SPDD  =  5300.00
STMRT=   6.4794 FLOWC= 45669.79 TEXL  =   311.41

ATTACHED GENERATOR CONDITIONS
-----------------------------------------------------------

TOTAL KW LOAD =   800.00
GENERATOR HORSEPOWER =  1184.41

GAS TURBINE CONDITIONS
-----------------------------------------------------------

SHP   = 15517.31 SPEED=      3600 TOUT  =    75.00 XIN   =     2.20
XOUT  =     9.50 HUMGR=   117.00 LHV   = 18120.00 T1    =   915.30
CP1   =   0.2712 SPFCS=   0.4549 WGAS  =406060.00

OVERALL FUEL RATE = 0.3370
--------------------------

TOTAL SHP = 21374.51
--------------------
```

```
CASE 5 -  60 PC POWER - 11/14/78 - J.H. - 5615 - 800 KW LOAD
--------------------------------------------------------------
HEAT BALANCE OUTPUT WITH INPUT STACK TEMP. - DUMPING
--------------------------------------------------------------

F1    = 42569.36 FTH   = 42322.36 FTG   =     0.00 FFP   =     0.00
FL    =   247.00 FEXST= 42103.04 GLAND=   219.31 FTGLN=     0.00
GL+VT=   319.31 FMU   =   247.00 FEVAP=   480.92 FDUMP=    52.93
FMCND= 43203.22 VENT  =   100.00 F2    =  6854.35 F1+F2= 49423.71
BLEED=     0.00 GPD   =  2061.64 WWPPH=     0.00

PDFT  =    25.00 PRES1=    25.00 PRES2=   368.00 PSH   =   360.00
PTH   =   354.60 PMCND=     1.68

TSH   =   855.00 TTH   =   850.00 TOFTL=   240.06 TEV   =   436.50
TEC   =   416.50 TDECO=    16.00 TBLED=     0.00 PINPT=    11.00
T2    =   447.50 TSTK  =   295.00 T1A   =   798.29 T2A   =   364.77

CP2   =   0.2563 CPSTK=   0.2482 CP1A  =   0.2637 CP2A  =   0.2530

HSH   =  1447.09 HTH   =  1444.66 SLEP  =  1038.67 HGLND=  1282.26
HTGLN=     0.00 HMCND=    63.36 HGLCF=    71.32 HDFTG=  1160.54
HDFTL=   208.41 HPR1L=   208.41 HPR1G=  1160.54 HFPX  =   209.83
HBLED=     0.00 HTECO=   395.19 HPR2L=   415.00 HPR2G=  1204.15

STEAM TURBINE CONDITIONS
--------------------------------------------------------------

SHPR  =  9000.00 SPDR  =  5500.00 SHPD  =  6428.35 SPDD  =  5500.00
STMRT=   6.5739 FLOWC= 42322.61 TEXL  =   304.74

ATTACHED GENERATOR CONDITIONS
--------------------------------------------------------------

TOTAL KW LOAD =   800.00
GENERATOR HORSEPOWER =  1184.41

GAS TURBINE CONDITIONS
--------------------------------------------------------------

SHP   = 13300.55 SPEED=     3600 TOUT  =    75.00 XIN   =     2.00
XOUT  =     8.40 HUMGR=   117.00 LHV   = 18120.00 T1    =   901.30
CP1   =   0.2704 SPFC8=   0.4793 WGAS  =382320.00

OVERALL FUEL RATE = 0.3508
--------------------------

TOTAL SHP = 18544.50
--------------------
```

```
CASE 6 -  50 PC POWER - 11/14/78 - J.H. - 5615 - [illegible]0 KW LOAD
--------------------------------------------------------------
HEAT BALANCE OUTPUT WITH INPUT STACK TEMP. - DUMPING
--------------------------------------------------------------

F1    = 39195.99 FTH   = 38967.99 FTG   =     0.00 FFP   =     0.00
FL    =   228.00 FEXST= 38750.53 GLAND=   217.45 FTGLN=     0.00
GL+VT=   317.45 FMU   =   228.00 FEVAP=   468.15 FDUMP=     4.28
FMCND= 39768.42 VENT  =   100.00 F2    =  6329.67 F1+F2= 45525.66
BLEED=     0.00 GPD   =  2006.90 WWPPH=     0.00

PDFT  =    25.00 PRES1=    25.00 PRES2=   338.00 PSH   =   331.00
PTH   =   326.90 PMCND=     1.61

TSH   =   847.00 TTH   =   842.00 TOFTL=   240.06 TEV   =   428.41
TEC   =   413.41 TDECO=    15.00 TBLED=     0.00 PINPT=    10.00
T2    =   438.41 TSTK  =   289.80 T1A   =   787.50 T2A   =   358.98

CP2   =   0.2558 CPSTK=   0.2475 CP1A  =   0.2651 CP2A  =   0.2525

HSH   =  1443.92 HTH   =  1441.51 SLEP  =  1041.71 HGLND=  1281.59
HTGLN=     0.00 HMCND=    61.96 HGLCF=    70.57 HDFTG=  1160.54
HDFTL=   208.41 HPR1L=   208.41 HPR1G=  1160.54 HFPX  =   209.75
HBLED=     0.00 HTECO=   389.66 HPR2L=   406.03 HPR2G=  1203.70

STEAM TURBINE CONDITIONS
--------------------------------------------------------------

SHPR  =  9000.00 SPDR  =  5500.00 SHPD  =  5805.52 SPDD  =  5500.00
STMRT=   6.6990 FLOWC= 38967.57 TEXL  =   297.91

ATTACHED GENERATOR CONDITIONS
--------------------------------------------------------------

TOTAL KW LOAD =   800.00
GENERATOR HORSEPOWER =  1184.41

GAS TURBINE CONDITIONS
--------------------------------------------------------------

SHP   = 11083.79 SPEED=     3600 TOUT  =    75.00 XIN   =     1.70
XOUT  =     7.30 HUMGR=   117.00 LHV   = 18120.00 T1    =   888.30
CP1   =   0.2696 SPFC8=   0.5131 WGAS  =356760.00

OVERALL FUEL RATE = 0.3645
--------------------------

TOTAL SHP = 15704.91
--------------------
```

APPENDIX C

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A DYNAMIC ANALYSIS OF A COGAS PROPULSION PLANT

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A Dynamic Analysis of a COGAS Propulsion Plant

by J. W. Abbott[1], J. G. McIntire[2], and C. J. Rubis[3]

ABSTRACT

A dynamic and control simulation of a 30,000 HP combined gas and steam (COGAS) propulsion plant powering a single screw controllable-pitch propeller frigate study ship is described. The propulsion plant consists of one General Electric LM 2500 gas turbine engine with a waste heat recovery boiler driving a steam turbine to augment gas turbine propulsion power. The all digital simulation includes equations and algorithms of the ship, propeller, and control system and dynamic models of the gas turbine and steam system. Results of computer simulations which investigated the performance of various COGAS plant control system designs are then presented. These simulations included acceleration from various forward speeds, crashback from full ahead, and crashahead from full astern. Results are described in terms of peak thrust and torque transmission loads plus ship maneuverability characteristics as compared to a simple cycle GT plant.

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I. INTRODUCTION

On January 1, 1974, the price of crude oil more than doubled. The subsequent shock wave resulting from this event has touched almost every area of economic and technical activity throughout the world. The maritime industry was certainly no exception and the impact of higher fuel costs has made itself felt to owners and operators of worldwide fleets whether they be commercial or naval. This impact is dramatically shown in Figure 1 which shows the cost leverage of fuel in a commercial operation of a 15,000 DWT Cargo ship as a function of days at sea.

Such economic factors have initiated several responses, including the one by many shipowners to simply "slow down" their ships at sea. However, this "solution" runs counter to past trends demanded by the shipping industry causing the gradual increase in ship speeds. Not only have ships speeds increased but ship size, leading to the growth in installed propulsion power reflected in Figure 2. Thus, the challenge to our technical community is to continue to meet the increase in power demand but with greater propulsion efficiency.

These problems, plus several other "mission oriented" objectives, have led the Navy into detailed studies of improved efficiency propulsion plants. Such considerations as endurance range, fuel tankage, power density and mission reliability are extremely important issues when designing a naval surface combatant. Many propulsion concepts optimize some of these parameters, only to yield unacceptable performance in the others (e.g., open cycle gas turbine propulsion yields a high power density only to provide moderate efficiency at cruise speed and below where 75% of at-sea operating time is spent).

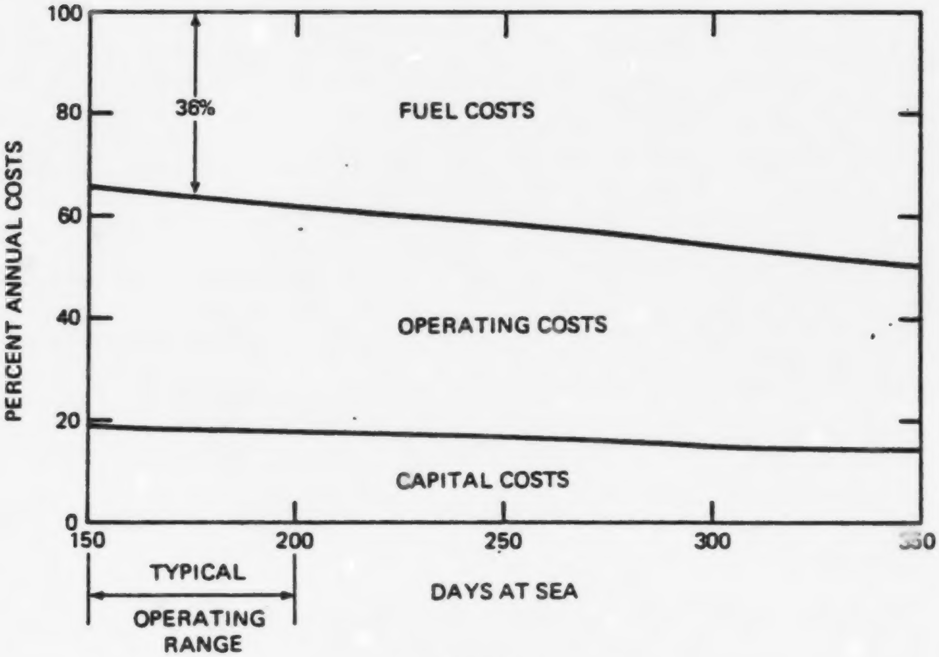

FIGURE 1

COST BREAKDOWN VS DAYS AT SEA

[FOR A 15,000 DWT CARGO SHIP]

SOURCE: REFERENCE [1]

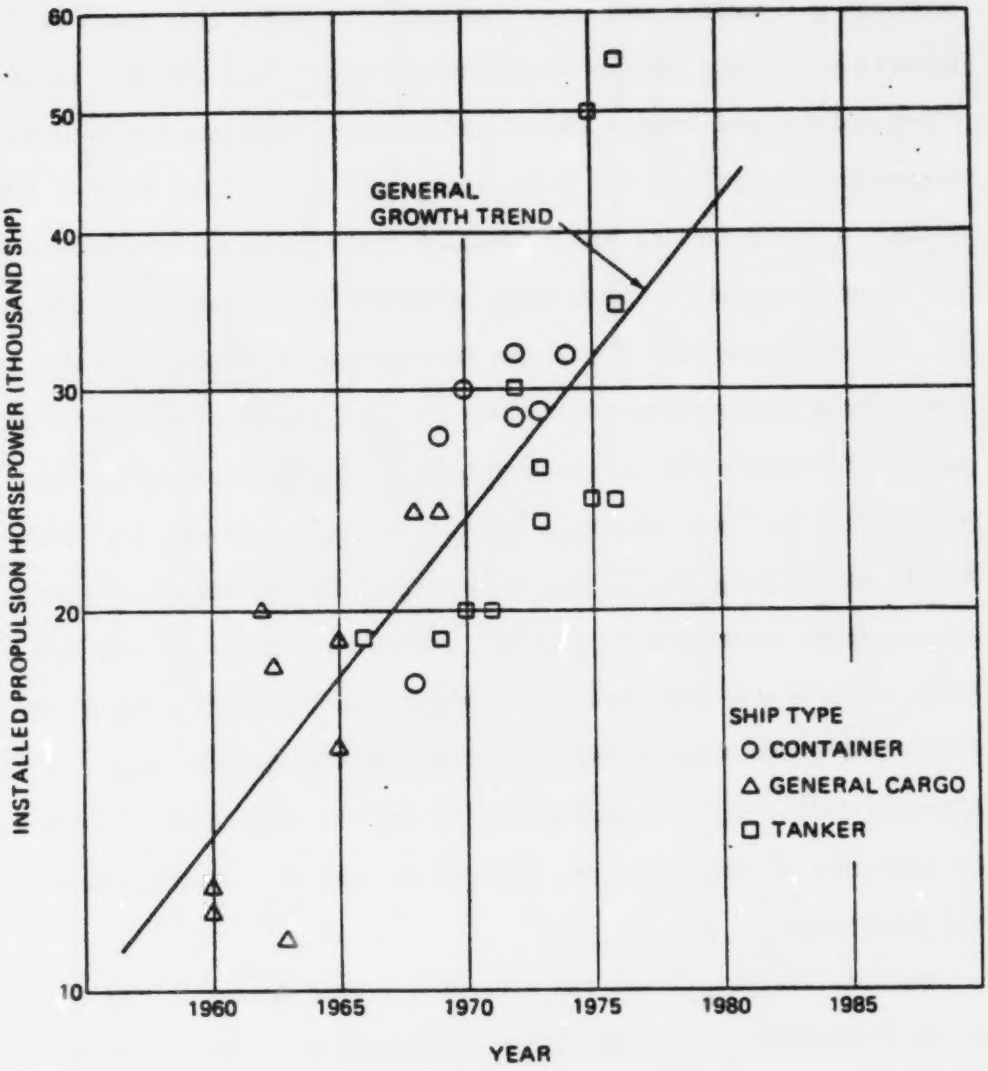

FIGURE 2
INSTALLED PROPULSION HORSEPOWER VS SHIP CONSTRUCTION YEAR

SOURCE: REFERENCE [2]

A propulsion concept which has the potential, not only to alleviate the fuel consumption problem, but to achieve Navy mission objectives as well, is a combined gas turbine and steam turbine plant which uses waste heat from the exhaust of the gas turbine to generate steam. This concept, called COGAS, (COmbined Gas and Steam turbine) makes use of both the Brayton and Rankine thermodynamic cycles. The plant contains a gas turbine engine, a waste heat boiler, a steam turbine and condensing system plus a conventional reduction gear, shafting and CP propeller. A schematic of the plant is shown in Figure 3. Power is generated in the gas turbine engine through direct conversion of fuel energy into mechanical torque (Brayton cycle). The exhaust gas (@ 940°F) is then routed through a heat exchanger (waste heat boiler) located in the stack where its temperature is lowered to 450°F. Steam is generated within the boiler at 600 psig (840°F) and subsequently fed to the steam turbine where it is converted to mechanical torque. The shaft outputs of both the gas turbine and steam turbine are connected to a common reduction gear which drives (at a reduced RPM) the ships propeller through an appropriate length of shafting. After expanding the 600 psig steam through the steam turbine, the Rankine cycle is completed in the normal way through the use of a vacuum condenser, deaerator and feed pumps.

Because the plant uses the otherwise wasted energy in the Brayton cycle, it is synergistic - that is it has a combined efficiency greater than the product of the two individual cycles. In effect, the steam portion of the combined cycles utilizes the "free energy" provided by the gas turbine exhaust and no additional fuel is required.* A comparison of COGAS thermodynamic efficiency with that of the straight Rankine and Brayton cycles is given in Figure 4.

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*There is a small fuel penalty (1%) caused by increased back pressure on the gas turbine due to the waste heat boiler. This is insignificant when compared to the 25% improvement in Fuel Consumption.

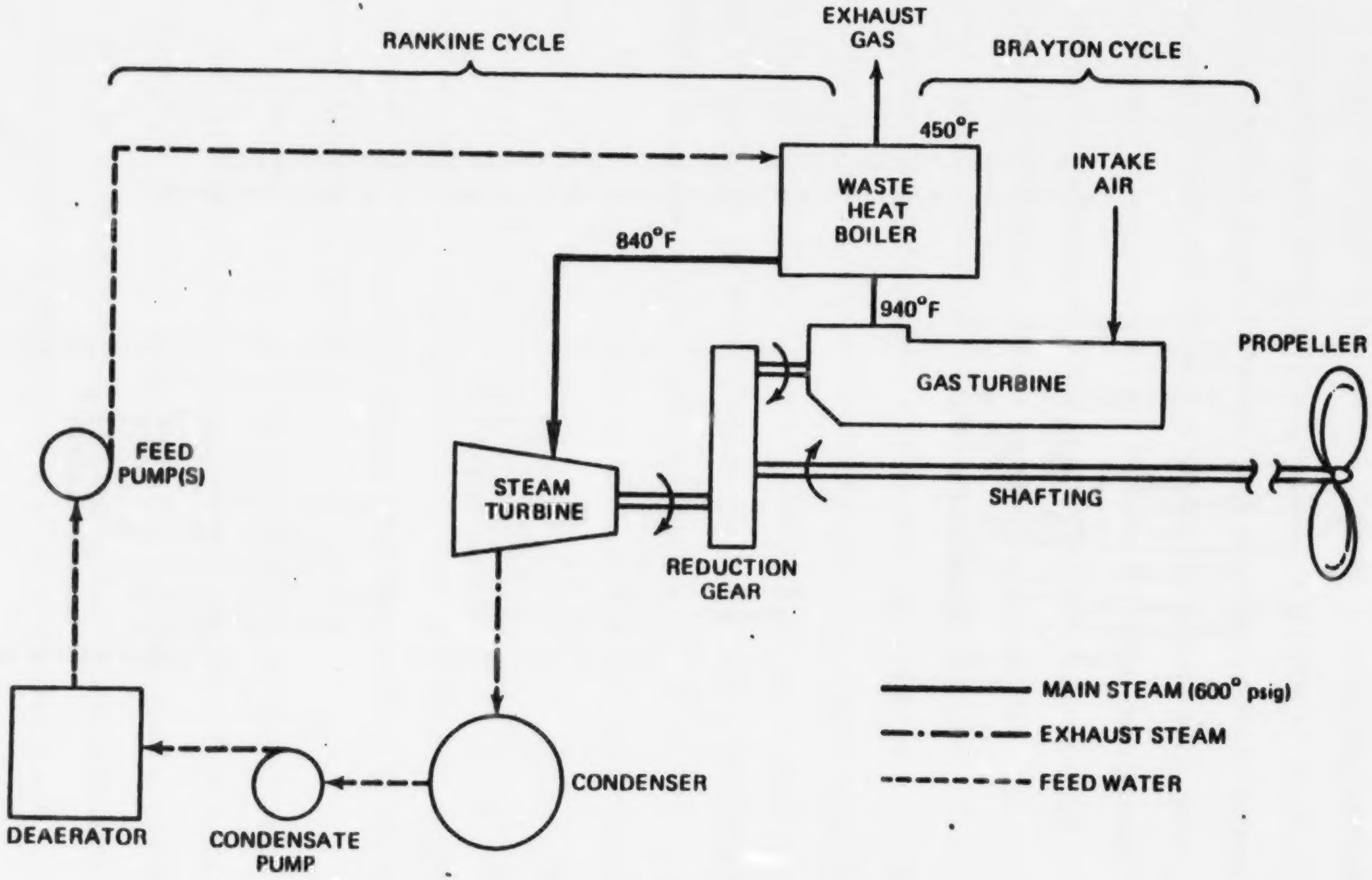
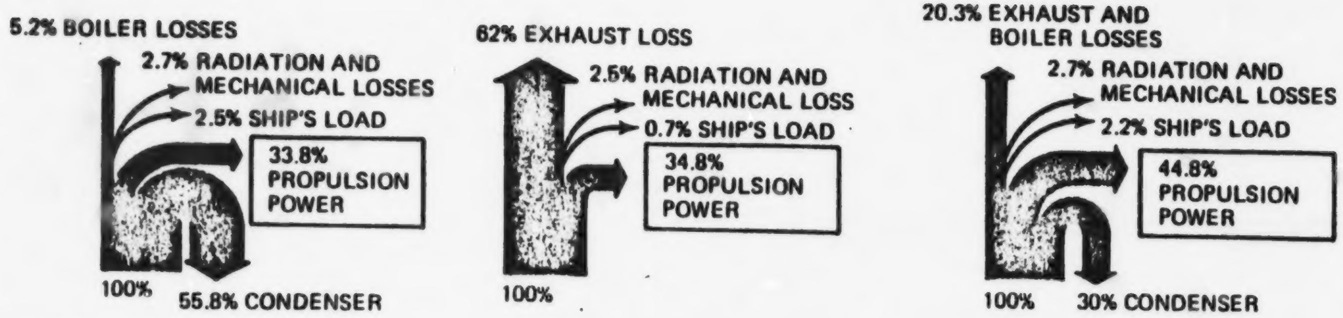

RANKINE CYCLE
EXHAUST GAS
BRAYTON CYCLE
450°F
WASTE HEAT BOILER
INTAKE AIR
840°F
940°F
PROPELLER
GAS TURBINE
FEED PUMP(S)
STEAM TURBINE
SHAFTING
REDUCTION GEAR
MAIN STEAM (600° psig)
EXHAUST STEAM
FEED WATER
CONDENSER
DEAERATOR
CONDENSATE PUMP

FIGURE 3
COGAS PROPULSION PLANT


5.2% BOILER LOSSES
2.7% RADIATION AND MECHANICAL LOSSES
2.5% SHIP'S LOAD
33.8% PROPULSION POWER
100%
55.8% CONDENSER
62% EXHAUST LOSS
2.5% RADIATION AND MECHANICAL LOSS
0.7% SHIP'S LOAD
34.8% PROPULSION POWER
100%
20.3% EXHAUST AND BOILER LOSSES
2.7% RADIATION AND MECHANICAL LOSSES
2.2% SHIP'S LOAD
44.8% PROPULSION POWER
100%
30% CONDENSER

(A) STEAM TURBINE PLANT (REHEAT)

(B) GAS TURBINE PLANT (OPEN CYCLE)

(C) COMBINED GAS AND STEAM TURBINE PLANT (COGAS)

FIGURE 4
COMPARISON OF THERMODYNAMIC EFFICIENCIES FOR STEAM (RANKINE), GAS TURBINE (BRAYTON) AND COMBINED CYCLE (COGAS) PLANTS.

Within the last three years a renewed interest has occurred regarding development of a COGAS propulsion plant utilizing an aircraft derivative second generation gas turbine. This has been true for several reasons - the most obvious one being a potential to dramatically reduce fuel consumption. Other, not so obvious, reasons which favor COGAS development for Naval use are; improved reliability over two shaft - single engine per shaft ships, improved part-load fuel consumption at cruise speeds, and the ability to fill the "power gaps" between available qualified marine gas turbines. These payoffs are gained with negligible penalties in manning, weight and volume. When the cost saving for fuel tankage structure (weight and volume) is considered, there is no penalty in ship acquisition cost. These and other factors were analyzed and presented in reference[3] which was based upon a 60,000 SHP plant (30,000 SHP/Shaft) designed for a 6000 ton destroyer.

This design was developed in great detail in a study sponsored by the Navy in 1973[4]. To develop 30,000 SHP/Shaft*, one LM 2500 was used in combination with a 600 psig steam turbine. Although this study developed the design of the waste heat boiler, selection of steam loop pressure, flow, and components, plus plant arrangement and weight values, it did not assess the dynamic response of the plant.

By the end of 1975, the need for COGAS in both naval and commercial applications was becoming apparent and the encouragement from within the technical community had reached a point where the investigation of control characteristics was felt to be mandatory. In particular the following questions were raised regarding control and dynamics of a COGAS propulsion plant.

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*Rated at 100°F standard day.

1. What is the complexity and extent of the control system required for a COGAS propulsion plant?
2. Given the control system, what are the characteristics of a COGAS plant for acceleration, deceleration, and steady state operation?
3. Are any special requirements associated with crashback or crashahead maneuvers?
4. What are the values of peak thrust and torque transmission loads of a COGAS plant as compared to a simple cycle GT plant for crashback, acceleration, and crashahead maneuvers?
5. Are there any special requirements associated with start-up and shut-down of a COGAS plant?
6. What are the general conclusions with regard to the ease or difficulty in controlling a COGAS propulsion plant?

To answer these questions a decade ago would have been next to impossible without building and testing an actual COGAS propulsion plant. However, in the late 1960's, work began to develop a new design tool-using the latest computer technology - to simulate (model) the complete dynamic response of gas turbine propulsion systems. Not by coincidence, this work in recent years centered around the LM 2500 since this engine was to be used for driving the Spruance class destroyers (DD-963) and later the FFG-7 class frigates. Some results from this work were presented at the 1972 SNAME Annual Meeting[5] and formed an excellent basis for developing an expanded model to include the COGAS configuration. This paper presents the results from such a COGAS simulation and indicates some very encouraging conclusions about controlling a COGAS plant.

II. PROBLEM STATEMENT

A. Ship and Propulsion System

The ship used for the COGAS propulsion plant dynamics investigation is a single screw controllable-pitch (CP) propeller 4000 ton frigate class study ship used in previous Navy simulations. This ship, the propulsion dynamics equations and methods of simulation are described in reference[5]. For the COGAS study ship, new 4 quadrant CP propeller data was used (including spindle torque data) and the propulsion plant was modified for a COGAS configuration.

The COGAS propulsion plant consists of one General Electric LM 2500 gas turbine engine with an exhaust heat recovery boiler driving a steam turbine to augment gas turbine propulsion power. Maximum power ratings and other conditions are as follows:

Gas turbine: $P_{ENG} = 26{,}390$ HP at $W_f = 10{,}000$ lbm/hr fuel rate,

$T_{AMB} = 59^{\circ}F$

Steam turbine: $P_{ST} = 7000$ HP

for gas turbine conditions of:

$W_8 = 143$ lbm/sec, $T_8 = 938^{\circ}F$

Combined Plant power: $P_S = (P_{ENG} + P_{ST})\, n_G = (26{,}390 + 7{,}000).97 =$ 32,390 SHP

Gas turbine and steam turbine (combined) speeds:

$N_E = 3600$ RPM, $N_{ST} = 5000$ RPM.

Steam plant conditions at steam inlet: temperature = $840^{\circ}F$, pressure = 615 psia.

The COGAS feasibility study[4] which preceded the dynamics simulation was conducted for design and sizing of the waste heat boiler and supporting systems and identifying the optimum configuration and major design parameters by a parametric evaluation involving steam pressure and temperature, component weights and volumes.

The primary purpose of this investigation was to study system dynamics and controls of a realistic COGAS plant in a typical naval ship application. Steam system design parameters used were those previously established in the original study while the gas turbine and ship model are those being used in other Navy ship dynamics and control simulations.

B. Steam Plant Description

The steam plant portion of the COGAS system is a Rankine Cycle powered solely by the waste heat of the gas turbine. A block diagram of the steam plant is shown in Figure 5. The steam plant equations (steady-state and dynamic) were written in a Fortran-like format to facilitate translation directly into a computer program. Steam plant parameters were designated by up to six alphanumeric characters. Station designations from 1 to 18 were used to identify various locations in the steam loop and gas path with the letter G (or S) after the number to indicate a gas (or steam) parameter. For example, T14S indicates steam temperature at station 14. The steam plant includes the following major components: waste heat recovery unit (WHRU), steam turbine, condensate system, deaerating feed tank (DFT), and feedwater system. The DFT functions as a surge tank, deaerates and heats the feedwater, and controls system water volume. Steam for DFT operation is extracted after the first stage of the seven stage impulse steam turbine. The steam turbine inlet flow is through a constant nozzle area and the turbine exhaust to the condenser which operates at a pressure of 3 in. Hg. The unfired WHRU consists of three sections: an economizer which heats the feedwater to a saturated liquid at the steam drum pressure, a forced circulation boiler, and steam drum producing saturated steam and a superheater which heats the steam to its final state exiting the WHRU.

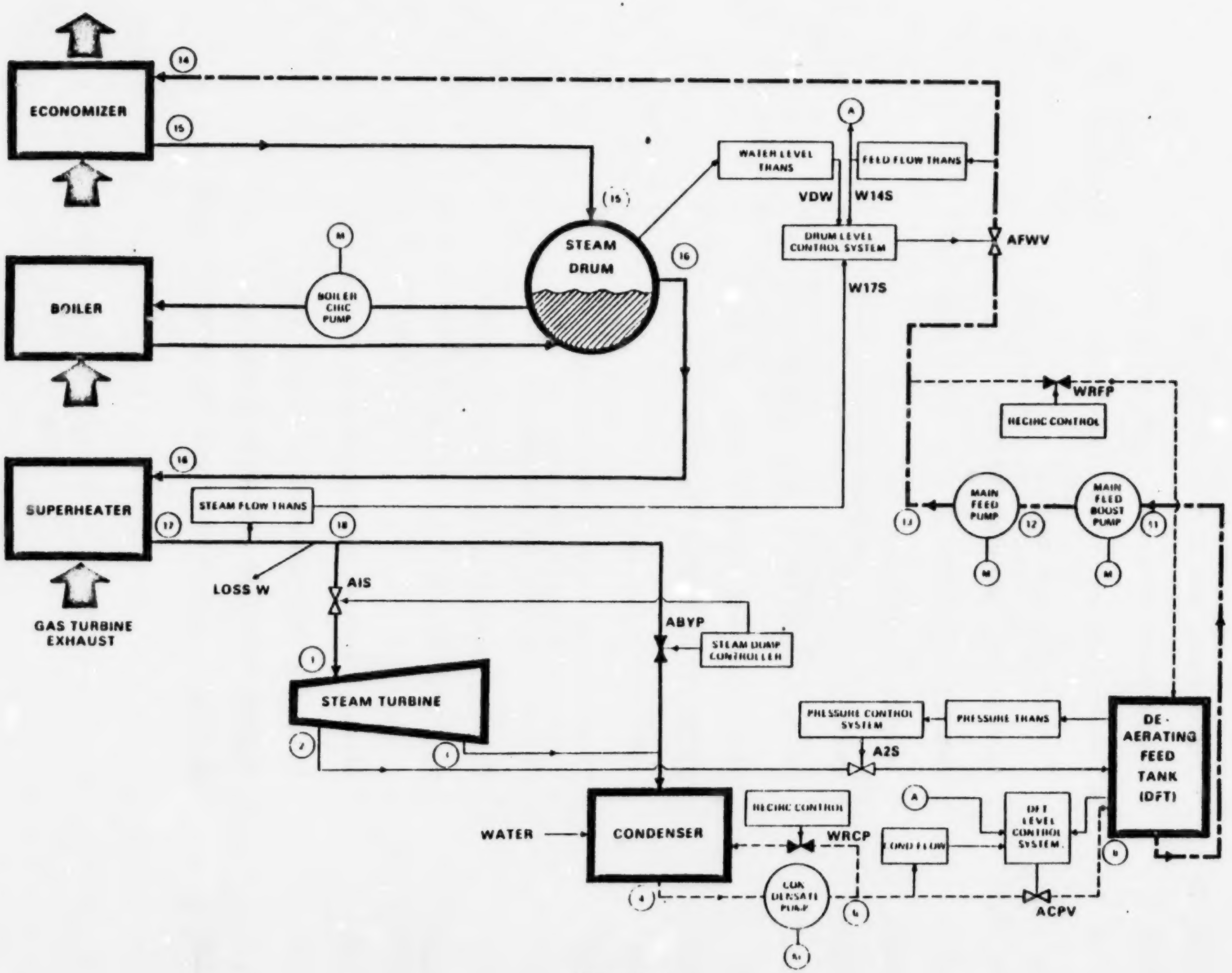

ECONOMIZER
BOILER
SUPERHEATER
GAS TURBINE EXHAUST
14
15
16
17
18
(15)
BOILER CIRC PUMP
M
STEAM DRUM
WATER LEVEL THANS
FEED FLOW THANS
A
VDW
W14S
DRUM LEVEL CONTROL SYSTEM
AFWV
W17S
WRFP
RECIRC CONTROL
MAIN FEED PUMP
MAIN FEED BOOST PUMP
13
12
11
STEAM FLOW THANS
LOSS W
AIS
ABYP
STEAM DUMP CONTROLLER
STEAM TURBINE
1
2
PRESSURE CONTROL SYSTEM
PRESSURE THANS
A2S
DE-AERATING FEED TANK (DFT)
WATER
CONDENSER
RECIRC CONTROL
WRCP
COND FLOW
DFT LEVEL CONTROL SYSTEM
CONDENSATE PUMP
ACPV
4
5
6

FIGURE 5
STEAM PLANT OF COGAS PROPULSION SYSTEM

T16G	Temperature, flue gas between superheater and boiler
T17G, T8	Temperature, gas turbine exit
T16S	Temperature, saturated steam in steam drum
T17S	Temperature, saturated steam at exit of superheater
T_8	Temperature, gas turbine exit
T8, T17G	Temperature, gas turbine exit
UA	Overall conduction of a section of the WHRU
UABLR	Overall conductance, boiler
UAECON	Overall conductance, economizer
UASH	Overall conductance, superheater
VD	Total volume of steam drum
VDW	Volume of saturated liquid in steam drum
VSH	Volume of superheater
WBYP	Flow of steam into bypass system
WD	Circulating flow in downcomers
W1S	Flow to inlet of steam turbine
W14S W15S	Feed flow through economizer
W16S	Steam flow entering superheater
W17S	Steam flow leaving superheater
W_8	Exhaust flow from gas turbine
XEXH	Quality of exhaust of ideal (isentropic) steam turbine

TABLE 1

DESIGN PARAMETERS FOR WASTE HEAT RECOVERY UNIT

ECONOMIZER • 4 pass, cross-counterflow • 3/4 in. O.D. finned steel tubes, 0.085 in. wall, 1.54 in. diameter fins (8.72/in.) • UA = 120,000 Btu/hr-°F at W8 = 128.5 lbm/sec
BOILER • 2 pass, cross-counterflow • 3/4 in. O.D. finned steel tubes, 0.085 in. wall 1.54 in. diameter fins (8.72/in.) • UA = 224,000 Btu/hr-°F at W8 = 128.5 lbm/sec
SUPERHEATER • 2 pass, cross-counterflow • 3/4 in. O.D. bare steel tubes, 0.085 in. wall • UA = 60,000 Btu/hr-°F at W8 = 128.5 lbm/sec

A simple steam plant control system with the steam plant "slaved" to the gas turbine was a design goal in this study. The WHRU extracts energy from the gas turbine exhaust and the steam turbine produces additional propulsion power. Except during start-up/shut-down and possibly during severe maneuvering transients, the inlet to the steam turbine is full-open. The steam turbine is assumed to operate under conditions of supersonic flow at all power levels, thus $W\sqrt{T/P}$ is always constant at the steam turbine inlet. The nature of the WHRU and fixed nozzle area turbine cause the thermodynamic cycle to shift as a function of gas turbine power level as indicated in Figure 8.

All pumps are electric motor driven. For minimum flow control the feed and condensate pumps recirculate back to the deaerating feed tank and condenser respectively.

The design parameters of the waste heat recovery unit are summarized in Table 1[3,4].

C. Integrated Throttle Control

The integrated throttle control for the gas turbine plant was designed with the utmost simplicity so as not to obscure the interpretation of transient loads and control problems in the COGAS mode of propulsion plant operation and still provide protection against excessive load levels. The entire propulsion plant is controlled via a single remote throttle lever. Propeller pitch and gas turbine power lever angle are ordered from a scheduler responding to remote throttle lever position. Steam plant controls are automatic, responding indirectly to the gas turbine power level command. The gas turbine plant propulsion control system is shown in Figure 6. The steady-state schedule between bridge throttle lever position (BTL), power lever angle (PLA) and propeller pitch ratio (PR) is also shown in Figure 6. This schedule was used in the simulation to order a pitch ratio and power lever angle for any bridge throttle command.

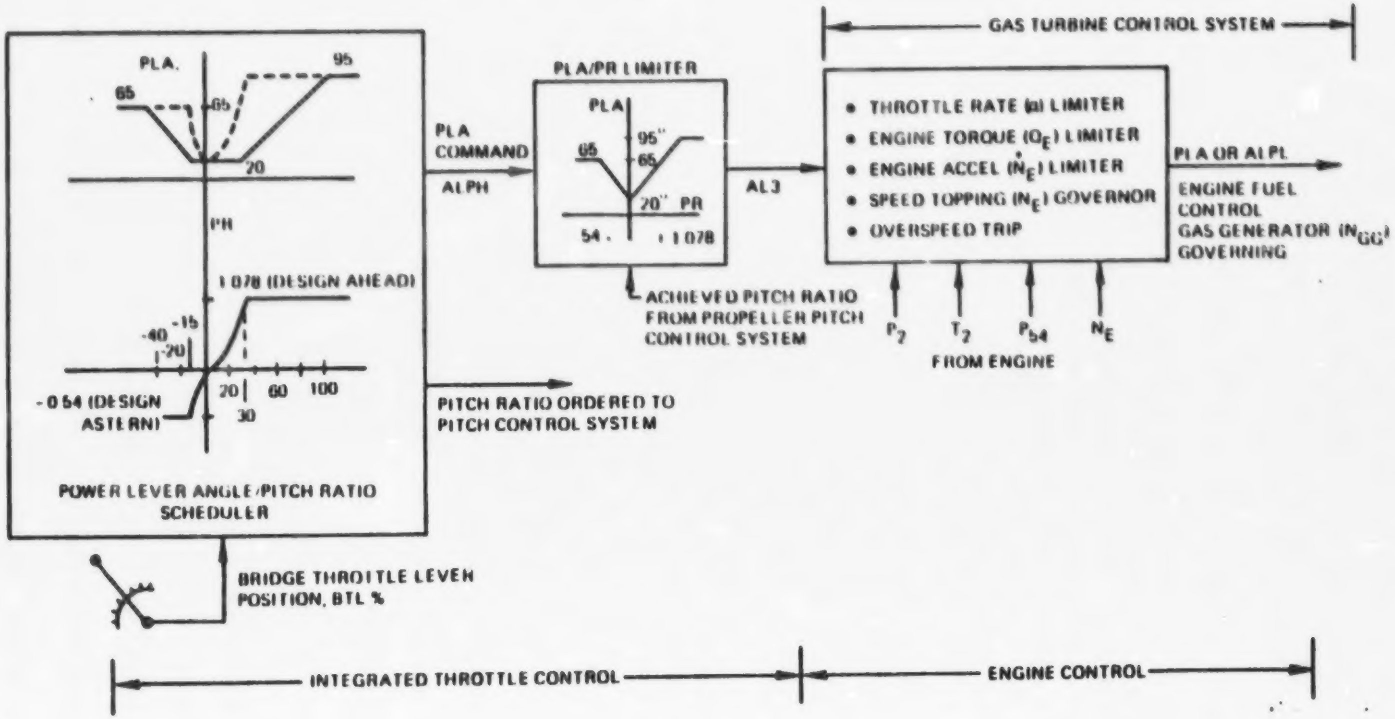

GAS TURBINE CONTROL SYSTEM
PLA.
65
65
95
20
PR
1.078 (DESIGN AHEAD)
-40
-15
-20
20
30
60
100
- 0.54 (DESIGN ASTERN)
POWER LEVER ANGLE/PITCH RATIO SCHEDULER
BRIDGE THROTTLE LEVER POSITION, BTL %
PLA COMMAND
ALPH
PITCH RATIO ORDERED TO PITCH CONTROL SYSTEM
PLA/PR LIMITER
PLA
65
95"
65
20"
PR
54.
+1.078
ACHIEVED PITCH RATIO FROM PROPELLER PITCH CONTROL SYSTEM
AL3
• THROTTLE RATE (α) LIMITER
• ENGINE TORQUE (Q_E) LIMITER
• ENGINE ACCEL ($\dot{N}_E$) LIMITER
• SPEED TOPPING (N_E) GOVERNOR
• OVERSPEED TRIP
P_2
T_2
P_{54}
N_E
FROM ENGINE
PLA OR ALPL
ENGINE FUEL CONTROL
GAS GENERATOR (N_{GG}) GOVERNING
INTEGRATED THROTTLE CONTROL
ENGINE CONTROL

FIGURE 6
GAS TURBINE PLANT PROPULSION CONTROL SYSTEM

A steady-state version of the ship propulsion simulation was developed for the purpose of steady-state performance evaluation. Using this steady-state simulation, the PLA/PR scheduler was designed to give a linear change in ship speed with bridge throttle lever position. There was nothing particularly critical in this choice of the PLA/PR schedule. The main purpose in this instance was to establish a reasonable schedule of PLA and PR and to compute several key steady-state parameters as a function of bridge throttle lever position, thereby establishing a frame of reference for the later dynamics simulations.

The PLA/PR limiter is a scheduler designed to limit peak transient loads by restricting PLA at low pitch ratios, thus preventing throttle commands that are excessive (in terms of peak propeller speeds, torques or thrusts) for any particular pitch ratio. The PLA/PR limiter (shown in dashed lines on Figure 6) is a propulsion maneuvering transient limiter which is inactive in the normal steady-state schedule. The operation of the PLA/PR limiter is as follows: Instantaneous propeller pitch ratio is used with the PLA/PR limiter characteristic to determine the maximum allowable value of PLA. If the commanded value of PLA exceeds the allowed value, then the PLA follows the limiter characteristics and the limiter is "active". If the commanded PLA falls below the limiter characteristics, the limiter has no effect and the limiter is "inactive."

D. Gas Turbine Controls

The gas turbine controls (fuel controls limiter for throttle rate, engine torque and engine acceleration, topping governor and overspeed trip) are those typically furnished with the LM 2500 engine. The engine throttle command (PLA) is translated to percent demanded gas generator speed ($\%N_{GG}$) which is an input to the fuel control system. A closed loop control maintains the speed of the gas generator (N_{GG}) at the demanded value. Such control is essentially power governing of the gas generator. The free turbine (engine) speed (N_E) is determined by a balance between the power produced by the gas generator and the power consumed by the load with no free turbine speed governing.

E. Steam Plant Controls

As previously noted, the steam plant is "slaved" to gas turbine exhaust conditions, with the main steam valve always open except during start-up, shut-down or possibly severe maneuvering transients. The steam drum level controller is a three element control based on drum level, feedwater flow rate and superheated steam flow rate. The deaerating feed tank (DFT) level controller is a three element control based on DFT level, condensate flow rate and feedwater flow rate. DFT pressure is maintained by steam extraction after the first steam turbine stage. Condenser hotwell level is maintained automatically using make-up-feed to or from a fresh water reserve tank. These control systems are shown in the steam plant diagram of Figure 5. Steam plant controls are discussed in greater detail later in the development of the steam plant mathematical models and simulation.

III. MATHEMATICAL MODELING AND SIMULATION

A. Ship and Gas Turbine Propulsion Plant Dynamics

A dynamics simulation of the ship and its gas turbine propulsion plant with associated control systems was developed around a complete dynamic simulation of the General Electric LM 2500 gas turbine engine.

This simulation with its CP propeller coupled thrust and torque equations was designed to permit any power level propulsion plant maneuver including acceleration, crashback, backing and crashahead. After the gas turbine plant model was completed and verified, the steam plant dynamic model was developed and these two models were then combined to become the COGAS ship dynamic model. The final COGAS ship dynamics simulation is quite complex, printing out a total of 160 different parameters every 0.1 seconds during maneuvering transients.

Results of gas turbine CP propeller dynamic ship simulations have been published by various authors especially in recent years. Therefore, gas turbine and ship dynamics are not discussed here. Rather, the emphasis is on the dynamic model of the steam plant because of its unique features (e.g., varying exhaust energy unfired boiler, and uncontrolled main steam throttle) together with the dynamics of the combined (COGAS) ship propulsion system.

B. Steam Plant Heat Balance

The steady-state heat balance simulation, developed as a stepping stone to the dynamic simulation discussed later, predicts more than 80 parameters specifying combined propulsion plant and ship performance at any given gas turbine power level. The heat balance simulation consists of: the WHRU model, the steam turbine model, and the condensate/feedwater systems which close the steam loop. Control systems are not explicitly included as part of the heat balance simulation; rather it is assumed the controls maintain proper conditions throughout the cycle. The following discussion provides a summary of the approach and some major assumptions used in the heat balance simulation.

For the waste heat recovery unit, it was assumed the heat transfer is governed by the gas side heat transfer coefficient, thus overall conductance, UA, can be written proportional to the 0.6 power of hot gas flow W_8, i.e.,

$$UA \propto W_8^{0.6}$$

Furthermore, it was assumed the steam entering the superheater is saturated vapor and the state of the feedwater entering the WHRU via the economizer is known. The WHRU design, including the UA data discussed earlier were taken from the original study[4], however the effectiveness was computed using the procedures of reference [6].

The heat transfer to the superheater QSH is calculated using the heat exchanger effectiveness method.

$$QSH = EFSH*CMIN*(T17G - T16S)$$

where

CMIN is the minimum hourly heat capacity rate of the gas and steam side, Btu/hr °F

EFSH is the superheater effectiveness

T17G and T16S are the temperatures of the gas turbine exit and saturated steam respectively

NOTE
* = computer symbol for multiplication

The superheater effectiveness can be calculated from the hourly heat capacity rates of the steam and gas sides of the superheater, the overall conductance of the superheater (UASH) and the number of passes in the superheater[6].

Heat transfer to the boiler QBLR is calculated using a log mean temperature difference.

$$\mathrm{QBLR} = \mathrm{UABLR}*(\mathrm{T16G} - \mathrm{T15G})/\mathrm{ALOG}(\mathrm{T16G} - \mathrm{T16S})/(\mathrm{T15G} - \mathrm{T16S}))$$

where

(T16G - T15G) is the gas temperature difference across the boiler and the other two terms represent temperature differences between gas and steam.

Unlike the superheater, the boiler model uses the calculated heat transfer to find a steam flow rate. The enthalpy change across the boiler and steam drum is h_{fg}, therefore the steam flow W16S leaving the drum is given by

$$\mathrm{W16S} = \mathrm{QBLR}/h_{fg}$$

For the economizer, the heat transfer is calculated using the heat exchanger effectiveness method

$$\mathrm{QECON} = \mathrm{EFE}*\mathrm{CMIN}*(\mathrm{T15G} - \mathrm{T14S})$$

where

CMIN is the minimum hourly heat capacity rate for the gas and feedwater, Btu/hr-$^{\circ}$F

EFE is the economizer effectiveness computed as before[6]

(T15G - T14S) is the difference in temperature of the gas entering the economizer and the feedwater entering the economizer

Like the boiler, the heat transfer rate to the economizer is used to calculate the steam flow through the economizer. This is accomplished by assuming that the flow leaving the economizer is saturated liquid and the flow entering the economizer is at the same enthalpy as the fluid in the DFT. Thus, the steam flow W15S through the economizer is

$$W15S = QECON/(H15S - H14S)$$

where

(H15S - H14S) is the feedwater enthalpy change across the economizer

At the steady-state design point W16S = W15S. While the method outlined above will produce values which differ slightly, the flow rate through the WHRU is taken as a weighted average of W15S and W16S. This technique accommodates the deviations of the flow exiting the economizer (W15S) from saturated liquid conditions. The error of this technique, under steady-state conditions is found to be small.

The computer implementation of the WHRU model is an iterative procedure. The model convergence is considered complete when the rate of convergence for flow, pressure and temperature falls below specified limits.

The steam turbine model responds to varying inlet steam flows and conditions and computes steam turbine power and steam conditions at the extraction and exhaust points of a simple pressure compounded impulse turbine. A seven stage turbine was selected using Robinson's Reheat Factor[7] and assuming an equal isentropic enthalpy drop across each of the turbine's seven stages. The model is based on the following assumptions: There is no throttle or nozzle area control. Turbine chest pressure will be that developed by the WHRU, no stage bypasses were included and choked flow exists in the turbine nozzles.

Using the fixed physical parameters of the steam turbine, off-design steady-state performance data is computed via an iterative procedure which determines stage available energy and stage efficiency. The steam properties in each of the stages is computed. The iteration is continued until the turbine exhaust pressure is within a stated tolerance of the condenser pressure.

A condenser pressure of 3 in. Hg abs is maintained by seawater circulation flow control and motor driven vacuum pumps over the entire steam plant operating range. Condensate pump minimum flow control is achieved by a recirculation line from the condensate pump discharge to the condenser hotwell which is assumed to be 2°F subcooled.[8]

A DFT pressure of 57 psia is maintained over the entire load range by controlling the extraction steam flow to the DFT. Feedwater pump minimum flow control is achieved by a recirculation line from the feedwater pump discharge to the DFT.

C. Steady-State Simulation Results

Table 2 presents steady-state performance data for the combined propulsion plant as a function of gas turbine fuel rate, W_f.

As the gas turbine fuel rate decreases, the gas exit temperature decreases then reaches a minimum at a low fuel rate and increases slightly as the idling level is reached. This temperature inversion due to degrading gas turbine efficiency at low gas turbine power levels (in the vicinity of 3000 HP) causes similar inversions of the steam temperature and enthalpy entering the steam turbine at this low power condition.

Figure 7 shows steam loop pressures and temperatures for various gas turbine fuel rates. Figure 8 is the steam loop thermodynamic cycle for two gas turbine power levels.

D. Steam Plant Dynamic Model

The dynamic response of the steam plant is represented by nine nonlinear, coupled differential equations describing the response of various components of the system. These differential equations are lumped parameter equations

TABLE 2

COGAS PROPULSION SYSTEM STEADY-STATE PERFORMANCE FOR DESIGN PITCH RATIO AND T_{AMB} = 59°F AT VARIOUS GAS TURBINE FUEL RATES

W_f lbm/hr	V kts	N RPM	P_{ENG} HP	P_{ST} HP	P_S HP	SFC $\frac{lbm}{HP \cdot hr}$	W8 lbm/sec	T8 °F	T16G °F	T15G °F	T14G °F	T1S °F	P1S psia	W1S lbm/sec
1000	8.10	65.1	350	418	687	1.456	27.2	722	649	335	318	706	97.6	2.51
2000	13.62	107.5	2164	1065	2999	0.667	54.4	676	627	379	352	661	156.8	4.11
3000	16.94	135.1	4729	1689	6047	0.496	74.1	686	640	407	374	669	207.0	5.40
4000	19.48	156.1	7551	2348	9400	0.426	89.1	719	669	433	391	696	261.4	6.74
5000	21.59	173.5	10465	3137	12983	0.385	100.1	767	708	458	405	732	323.4	8.21
6000	23.27	188.0	13385	3978	16634	0.361	109.1	816	747	482	418	767	388.0	9.71
7000	24.70	200.6	16406	4786	20359	0.344	117.7	854	779	501	428	792	448.3	11.12
8000	26.07	212.3	19626	5609	24300	0.329	125.7	890	807	520	437	813	509.0	12.53
9000	27.19	222.5	22805	6393	28167	0.320	133.6	920	831	536	445	828	566.4	13.88
10,000	28.13	231.8	26221	7075	32177	0.311	142.1	939	847	549	453	836	616.6	15.09

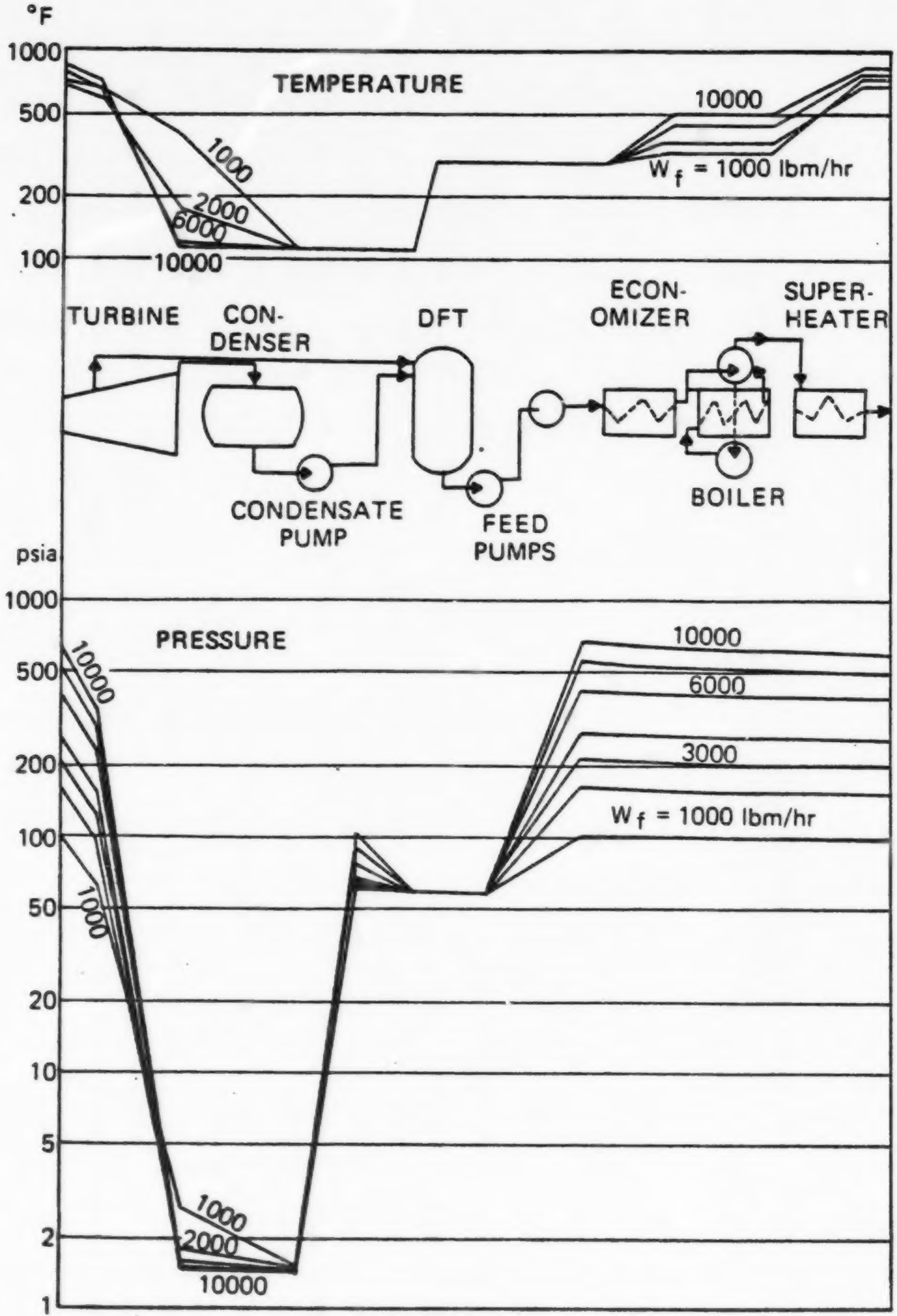


FIGURE 7
STEAM LOOP PRESSURES AND TEMPERATURES
FOR VARIOUS GAS TURBINE FUEL RATES

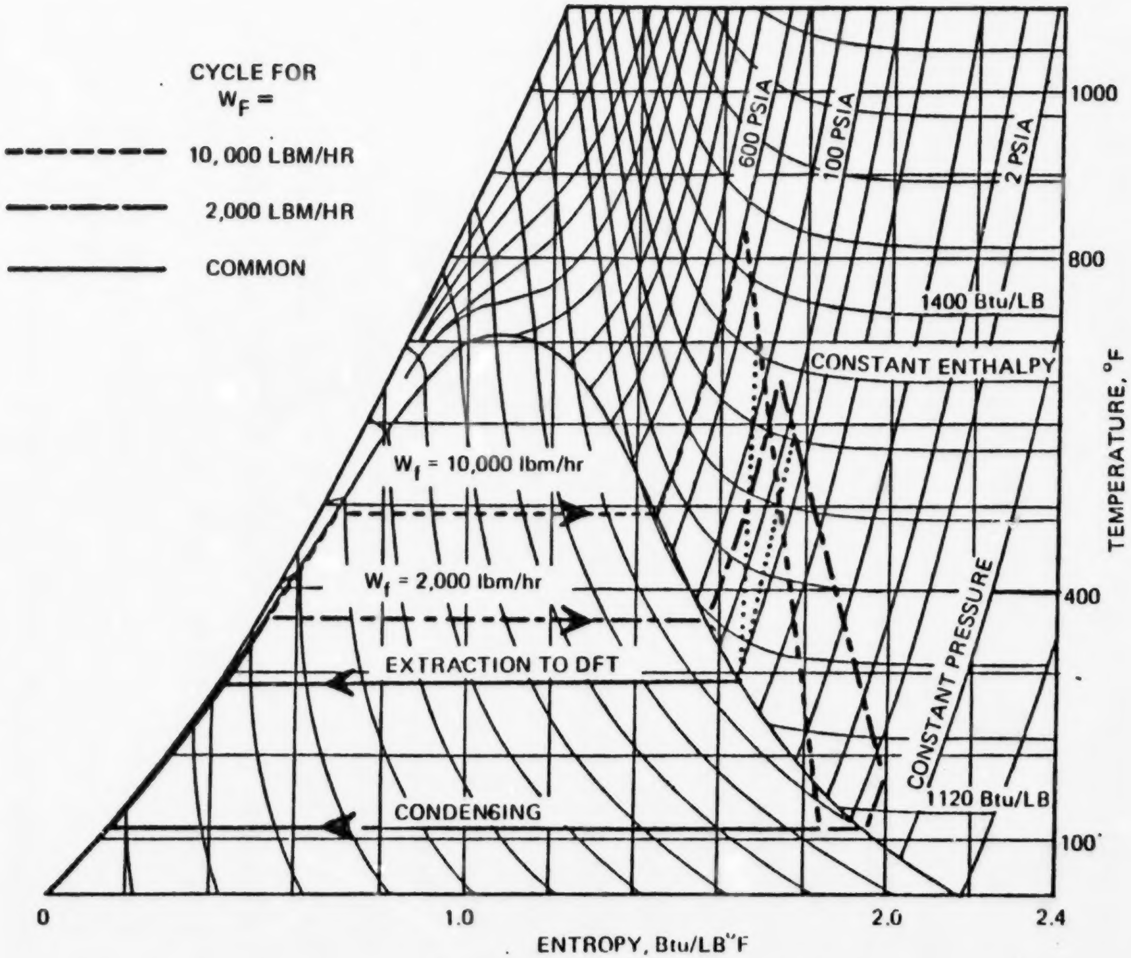


FIGURE 8

STEAM PLANT THERMODYNAMIC CYCLE FOR VARIOUS GAS TURBINE POWER LEVELS involving the conservation of mass and energy at various locations in the steam loop. The lumped parameter approach rather than a much more complex distributed parameter representation requiring many more differential equations was used. This approach can provide a good model for control system preliminary designs and investigations of ship and plant dynamic response during maneuvering transients while still retaining reasonable size.

The simulation includes dynamic models for the WHRU, the turbine and for each of the three major steam plant control loops: the DFT level control, the DFT pressure control and the steam drum level control. Minimum flow control for the condensate and feedwater pumps and condenser hotwell level is assumed ideal. Steam properties throughout the cycle in the simulation are based on the ASME steam tables. The following is a summary description of the steam plant dynamic model including the steam plant control systems.

- Condenser/Hotwell

  The condenser and hotwell are modeled as two separate components. The condenser is assumed ideal, it operates at a constant pressure of 3 in. Hg abs. and cools the steam turbine exhaust to an enthalpy corresponding to saturated liquid at this pressure. The hotwell is modeled as a mixing tank for the condenser flow, bypass flow, recirculation flow and make-up-feed flow using mass and energy balance differential equations.

- Condensate System

  The condensate system model includes the piping from the condenser to the DFT, a motor driven condensate pump, a recirculation system to the condenser for minimum flow control, a condensate flow control valve for DFT level control, and an extraction flow control valve and piping for DFT pressure control. The piping loss is assumed to be proportional to flow squared. The flow control valve pressure drops are assumed proportional to flow squared and inversely proportional to area.

- Deaerating Feed Tank (DFT)

  The deaerating feed tank is designed to serve several plant functions among which are the removal of oxygen and non-condensable gases from the condensate and the preheating of the condensate. The DFT model is developed assuming a single section, lumped parameter model sufficiently represents the important dynamic effects. Thus, the DFT is represented by conservation of mass and energy differential equations.

- Steam Generator Feedpump System

  The steam generator feedpump system model includes the piping from the DFT to the economizer, constant speed motor driven booster and feedpumps, a recirculation system to the DFT for minimum flow control and a feedwater flow control valve. Piping loss and flow control valve pressure drops were modeled as for the condensate system.

- Economizer

  The economizer model is developed based on the following assumptions: gas side dynamics are fast and can be considered algebraic, heat transfer is mostly convective and depends primarily on the gas side (although steam side coefficient effects were also investigated).

Using the heat exchanger effectiveness method the heat transfer rate of the economizer is

QECON = EFE*CMIN*(T15G - T14S)

where

EFE is the effectiveness of the economizer[6]

and

CMIN is the minimum heat capacity of the gas side and the feedwater side

The gas side temperature leaving the economizer is given by

T14G = T15G - QECON/(W8*CP15G)

where

CP15G is the specific heat of air at the gas side entrance of the economizer

The energy balance for the water side of the economizer yields

$$\frac{d}{dt}\text{ (H15S) = DH15S =}$$

(QECON - W14S*(H15S - H14S)/MECON

where MECON is the mass of the economizer.

- Downcomers

Feedwater from the economizer section is supplied to a perforated distribution pipe located at the bottom of the steam drum. The downcomers, which deliver the fluid to the circulating pump suction, are modeled assuming there is no mass storage in the downcomers and the total circulating flow is constant.

The energy relationship for adiabatic mixing of the feedwater flow and flow from the steam drum is

HD = ((WD - W14S)*HDW + W14S*H15S)/WD

where

HDW is the steam drum water enthalpy,

HD is the enthalpy of the water entering the downcomers, and

WD is the constant circulating flow rate through the boiler which is approximately five times the full power superheater flow level

- Boiler

Feedwater discharged from the boiler circulating pump rises in tube circuits and is subject to heat transfer from the hot gas before reentering the steam drum as a water-steam mixture. The energy balance equation for the water-steam mixture leaving the boiler is

$$\frac{d}{dt}\,(HW) = DHW = (QBLR - WD(HW - HD))/MBLR$$

where

HW is the enthalpy of the steam water mixture leaving the boiler,

QBLR is the heat transfer to the water in the boiler, and

MBLR is the boiler mass.

The heat transfer relationship for the boiler is based on the same assumptions as the economizer. The gas temperature leaving the boiler, T15G, is calculated using a logarithmic temperature difference and heat transfer through the boiler is written as

$$QBLR = (T16G - T15G)*W8*CP16G$$

where CP16G is the specific heat of air at the gas side entrance to the boiler.

- Steam Drum

The steam drum is an integral part of the circulating loop which contains the downcomer and boiler tube circuits. Incoming feedwater mixes with saturated liquid from the drum and enters the downcomers while a liquid-vapor mixture enters the steam drum from the boiler. Saturated steam passes through scrubbers designed to remove liquid before it leaves the steam drum and enters the superheater. An equilibrium model is assumed for the steam drum with the steam and water temperatures equal to the saturation temperature corresponding to the current steam drum pressure.

Mass and energy balances for the steam drum are

$$\mathrm{W14S} - \mathrm{W16S} = \frac{d}{dt}\,(\mathrm{VD} - \mathrm{VDW})*\mathrm{R16S} + \mathrm{VDW}*\mathrm{RDW}$$

and

$$\mathrm{WD}*\mathrm{HW} - (\mathrm{WD} - \mathrm{W15S})*\mathrm{HDW} - \mathrm{W16S}*\mathrm{H16S}$$
$$= \frac{d}{dt}\,(\mathrm{VD} - \mathrm{VDW})*\mathrm{R16S}*\mathrm{H16S} + \mathrm{VDW}*\mathrm{RDW}*\mathrm{HDW}$$

where VD is the total drum volume and VDW, RDW, HDW are respectively the volume, density and enthalpy of the saturated liquid in the drum.

The dynamics of the steam drum are described by $\frac{d}{dt}$ (R16S) and $\frac{d}{dt}$ (VDW) derived from the above differential equations.

- Superheater

The fluid in the boiler and economizer is assumed incompressible, however, for the fluid leaving the steam drum this assumption is no longer valid. In the mathematical model of the superheater a single section lumped parameter representation has been used with the total pressure drop assumed to take place before entering the heat transfer section. Momentum, mass and energy equations are written for compressible flow.

$$P16S - P17S = FSH*W16S**2/R16S$$

$$\frac{d}{dt}(R17S) = (W16S - W17S)/VSH$$

$$\frac{d}{dt}(H17S) = (QSH - W16S*(H17S - H16S))/MSH$$

where FSH, VSH, and MSH are respectively the flow coefficient, volume and mass of the superheater.

The heat transfer and energy balance equations for the gas side are based on the same assumptions as were used for the boiler and economizer.

$$QSH = KSH*(W8/KW8)**(T17G - T17S)$$

$$T16G = T17G - QSH/W8/CP8$$

where T8 is the gas side turbine exhaust temperature and KSH is the heat transfer coefficient for the superheater.

- Turbine and Bypass System

The steam turbine is intended to develop as much power as possible from the steam made available by the waste heat recovery unit. The turbine, comprised of seven impulse stages, is modeled as an isentropic nozzle with an ideal gas flow at sonic velocity at the throat. The main steam valve (A1S) closes and the dump valve opens, dumping steam directly to the condenser hotwell. Steam flow to the turbine and to the bypass system are modeled in a similar fashion.

$$W1S = K1S*A1S*P17S/SQRT(T17S + 460.)$$

$$WBYP = K1S*ABYP*P17S/SQRT(T17S + 460.)$$

where

K1S is the flow coefficient for the main steam valve,

A1S is the normalized effective flow area of the main steam valve, and

ABYP is the normalized effective flow area of the bypass system valve

The steam and bypass valve operation is constrained such that

A1S + ABYP = 1.

By assuming the extraction pressure is proportional to the flow in the turbine downstream from the extraction point, and sonic flow exists in the turbine, the state at the extraction point can be calculated. The extraction flow is controlled by valve A2S which responds to the DFT pressure controller.

Assuming the quality of the exhaust steam of an isentropic turbine is the same as the quality of the exhaust steam for the real turbine, the quality of the exhaust is taken as

XEXH = (S17S - SF)/SFG

where SF and SFG are the entropy of liquid at 3 in. Hg and the increase of entropy during vaporization at 3 in. Hg, respectively.

The dynamic steam turbine efficiency, EFF, is calculated from the steam flow rate based on the steady-state model of steam turbine efficiency. The exhaust quality determines the isentropic exhaust enthalpy which is used with the steam turbine efficiency and inlet enthalpy to compute the enthalpy of wet steam leaving the turbine.

H3WS = H17S - EFF*(H17S - HEXH)/100.

The external horsepower is computed as:

PST = ETAST*(W1S*(H17S - H2S)
+ W3S*(H2S - H3WS))*3600./2545.

- Steam Plant Control Systems

The simulation includes dynamic models for each of the three primary control systems as discussed below:

Water level within the steam drum is controlled by adjustment of the effective flow area of the feedwater valve (AFWV), using a three element level controller. For volume changes less than 15%, an approximation of volume based on vertical sides in the cylindrical drum is within 0.25%. Thus, for simplicity, the drum water volume rather than water level is used as the control signal. In addition to the drum water volume set point, KVDW, the controller senses: economizer flow rate, W14S, flow exiting the superheater, W17S, and drum water volume, VDW. The drum water volume is maintained at or near the set point by forcing W14S to correspond to W17S and providing volume trim.

The proportional-plus-integral controller responds to the drum error signal, and determines the desired feedwater valve position, AFWVT. The desired valve area, AFWVT, is limited to fall within the acceptable valve operating range and a first-order lag represents the valve area response.

[illegible]

The DFT level control is a three element controller similar to the drum level controller with a first-order lag representing valve area response.

The operating pressure of the DFT is controlled by adjustment of the steam extraction flow control valve area, A2S, with the proportional controller responding to the DFT pressure error signal and a first order lag to represent valve area dynamics.

IV. RESULTS OF DYNAMIC SIMULATION

To investigate the dynamic relationships which exist between the gas turbine and steam plants, various ship maneuvers were simulated. Figure 9 illustrates the dynamic response for the combined propulsion plant during a crashback maneuver from maximum ahead to maximum astern power. The throttle lever, is linearly ramped from full ahead to full astern in one second. The power lever angle, PLA, is ordered to about 20° corrresponding to an idling fuel rate. Gas turbine fuel reduction begins immediately and reaches the ordered idling value in less than 5 seconds. Coincident with the ordered reduction in PLA, the pitch control system orders pitch ratio to its design astern value (-0.54). Pitch reduction begins immediately at a rate determined by the pitch stroke time. When the decreasing pitch ratio reaches zero, the PLA is commanded to the maximum astern power position and an increase in fuel flow to the gas turbine begins, reaching the final fuel rate in about 10 seconds. Pitch ratio continues to decrease until the design astern value is achieved where it remains for the duration of the crashback maneuver. Turbine fuel, torque and speed increase, reaching their final values with the ship moving astern.

During the crashback maneuver, steam plant response is determined by gas turbine exit conditions and steam plant controller actions. Steam turbine power decays slowly due to the large thermal lag inherent in the WHRU. Changing gas turbine exit conditions cause reduction in the WHRU steam generation capability. Decrease in the flow rate of steam exiting the superheater causes a corresponding decrease in feed flow to the economizer to maintain steam drum level. Steam drum level controller action is to close the feed water flow control valve (AFWV). Reduction in feed flow initiates DFT level controller

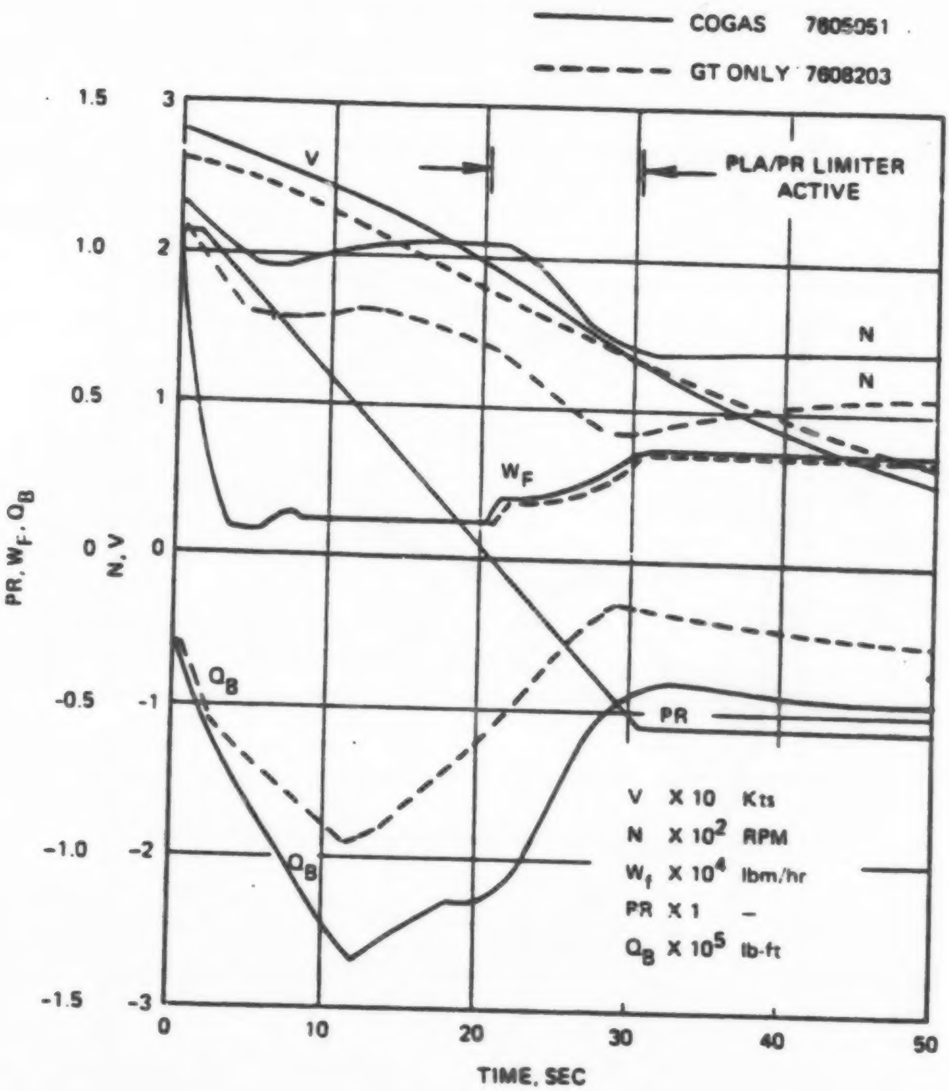


FIGURE 9, a
COMPARISON OF PEAK LOADS FOR GAS TURBINE AND COGAS PROPULSION PLANTS

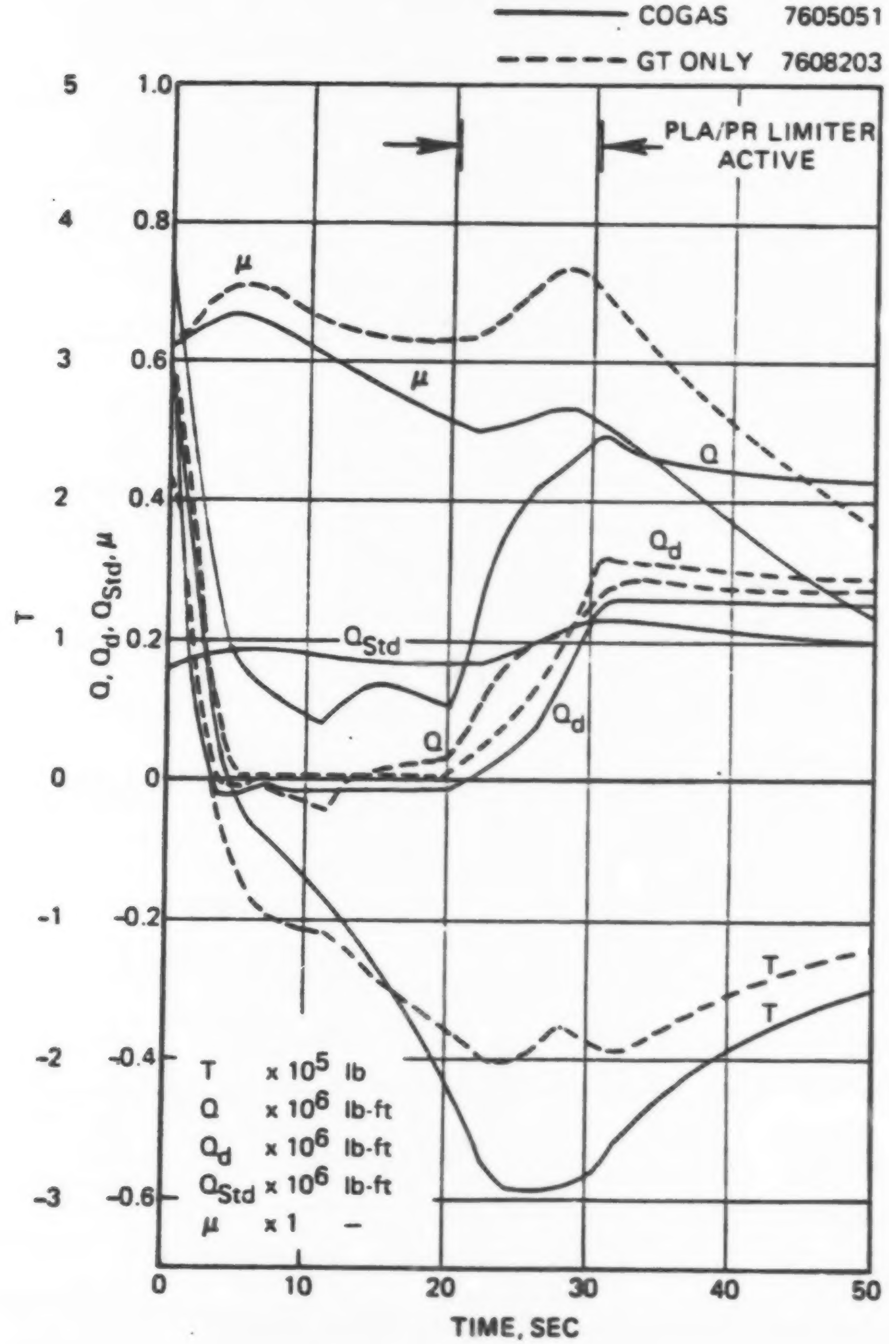


FIGURE 9,b
COMPARISON OF PEAK LOADS FOR GAS TURBINE AND COGAS PROPULSION PLANTS

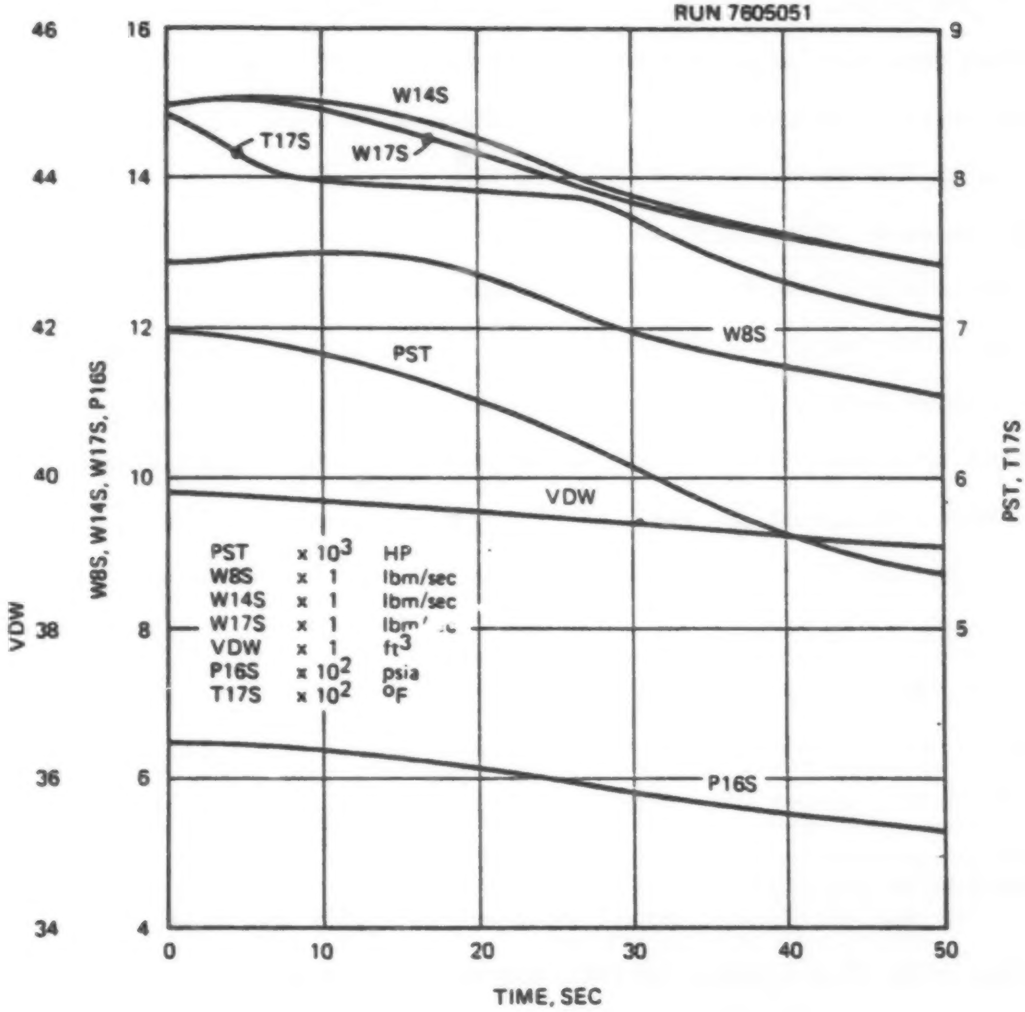


FIGURE 9, c
STEAM PLANT PARAMETERS DURING COGAS CRASHBACK

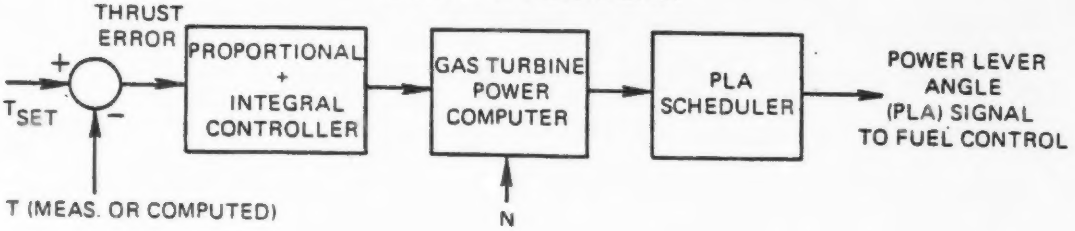
action to close down tne condensate flow valve (ACPV) to maintain DFT level with acceptable bounds. The DFT pressure (and hence its control valve area, A2S) remain practically constant during the first 50 seconds of the maneuver due to the large mass of water contained in the DFT.

The propeller "watermilling" or speed-increase hump occurs soon after pitch reduction begins and is characteristic of CP propeller crashback maneuvers. Comparison of a COGAS crashback with a gas turbine-only crashback is also shown in Figure 9. Although almost 5000 HP is still provided by the steam turbine even at zero pitch ratio, the propeller speed-up was much less than expected even with no steam dump. One reason was that at zero pitch ratio, this propeller is still absorbing 14% of full-ahead torque.

Other factors which tend to decrease the speed hump are the much increased drive train inertia due to the steam turbine and the lowered propeller modified advance coefficient (μ) which in this case increased propeller torque compared to a gas turbine-only crashback. For example, in the COGAS crashback (run 7605051) at the peak of the speed hump occurring at 20.9 seconds (where $\dot{N} = 0$, $PR \cong 0$) the propeller absorbs 82% of the steam turbine power while the sum of drive train friction and windmilling losses of the idling gas turbine's free turbine absorb 18%. Therefore, for the propeller used, steam dump proved unnecessary to control overspeed.

The peak thrust load during crashback is higher due to the additional steam power of the COGAS plant and the increased speed hump. To decrease the peak thrust and to maintain a large thrust over the whole crashback maneuver, several crashback optimization controllers were investigated. One such control scheme and results of a crashback maneuver are shown in Fig. 10. In this case, the pitch rate was decreased starting at PR = 0 to decrease the peak

CRASHBACK DYNAMIC THRUST CONTROLLER
COGAS PROPULSION PLANT


THRUST ERROR
+
−
T_{SET}
PROPORTIONAL + INTEGRAL CONTROLLER
GAS TURBINE POWER COMPUTER
PLA SCHEDULER
POWER LEVER ANGLE (PLA) SIGNAL TO FUEL CONTROL
T (MEAS. OR COMPUTED)
N

STOPPING PERFORMANCE FOR THRUST CONTROLLER

	NORMAL	THRUST CONTROLLER	% IMPROVEMENT
T_{PEAK} (LB)	−297150	−262851	11.5
T_{STOP} (SEC)	71.9	59.1	17.8
S_{STOP} (ft)	1498	1430	4.5

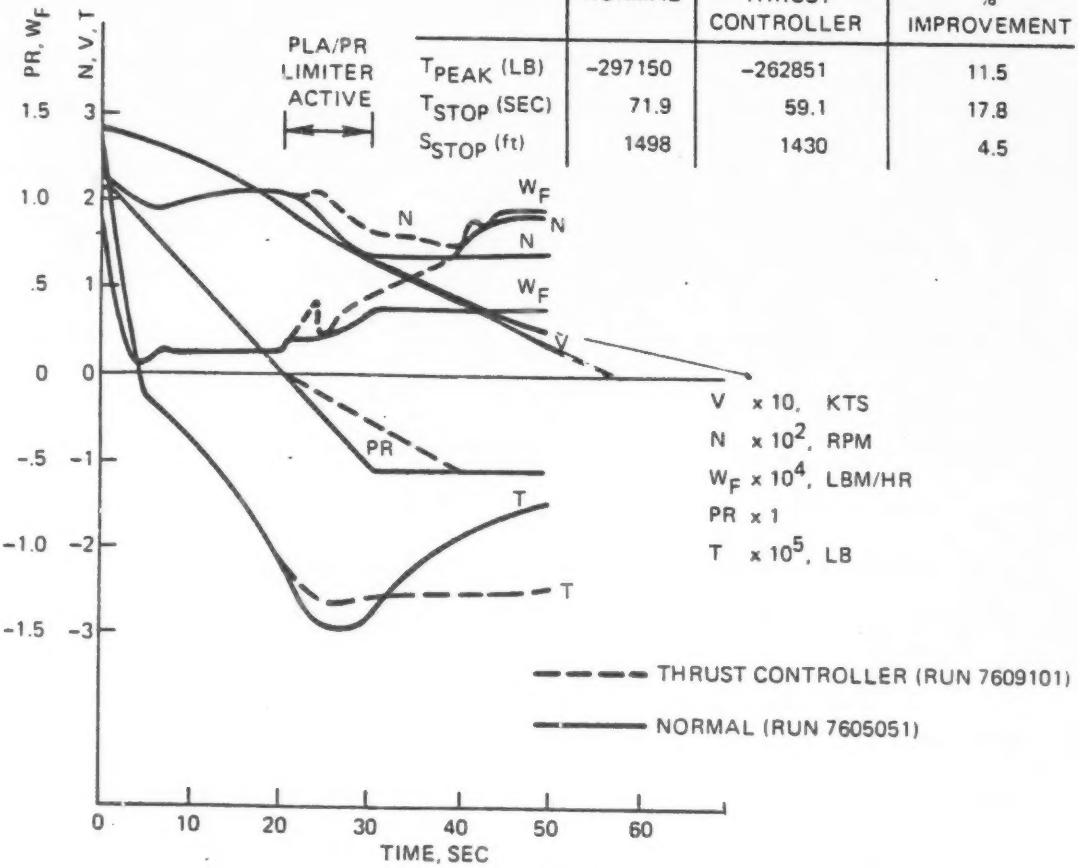
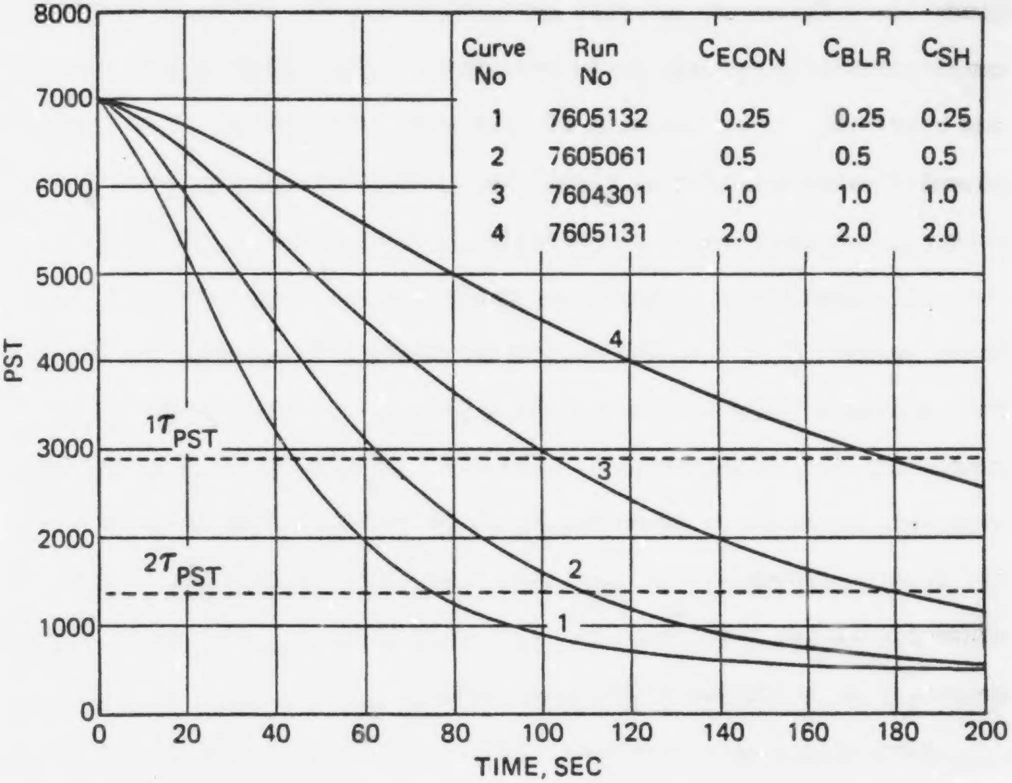


FIGURE 10
COGAS CRASHBACK WITH PITCH RATE CHANGE
AND DYNAMIC THRUST CONTROLLER

thrust, and a closed loop thrust controller used to maintain a large constant negative thrust. From the performance table, it can be seen that a simultaneous reduction in propeller peak thrust and ship head reach was obtained by this control sequence improvement.

Results of crashahead to full ahead from full astern and acceleration to full ahead from dead-in-the water were dominated by the gas turbine and its control systems. In these maneuvers, the gas turbine torque limiter and the PLA/PR limiter were constraining gas turbine PLA.

To investigate the effect of WHRU mass on steam plant response, a series of power chops were simulated using the steam plant dynamic model. These power excursions involve a time zero step decrease in gas turbine exit flow and temperature conditions from full power to idle. The mass of each section of the WHRU (economizer, boiler, superheater) is multiplied by a constant scale factor (C_{ECON}, C_{BLR}, C_{SH}). Using these mass scale factors, investigations were made to determine the change in steam plant response with variations in WHRU mass. No control system adjustments were made to compensate for changes in the systems response. The steam plant power response "time constants"(1_{PST}, 2_{PST}) correspond to the time required for the steam turbine power, P_{ST}, to achieve 63% and 87% of the change from full to idle power respectively. While these times are not true time constants (the system is very complex and nonlinear) they do provide a measure of the response rate. Figure 11 illustrates the variation in power response for mass changes in all section of the WHRU by factors of 0.25, 0.5, 1.0, and 2.0. As expected, a reduction in WHRU mass corresponds to a more rapid response and vice versa. For these various mass factors, there is no significant difference in the shape of the power curve.



Curve No	Run No	C_{ECON}	C_{BLR}	C_{SH}
1	7605132	0.25	0.25	0.25
2	7605061	0.5	0.5	0.5
3	7604301	1.0	1.0	1.0
4	7605131	2.0	2.0	2.0

FIGURE 11
THE EFFECT OF VARIATION IN WHRU MASS ON STEAM PLANT POWER RESPONSE

V. SUMMARY AND CONCLUSION

Over the last three years, a continuous effort has been made to establish the feasibility of a COGAS plant on naval and commercial ships. The original study[4] and follow-on analyses have addressed the issues of plant configuration, component sizing, steady state performance, fuel consumption, plant efficiency, and operating costs. Results of this work have been quite encouraging and have indicated a number of significant advantages of the COGAS plant over a conventional gas turbine plant design for certain ship applications.

This paper begins where the original study[4] left off and presents the investigation of one of the last major unresolved issues; the feasibility of controlling a COGAS plant, and the impact such a control system has on plant complexity and ship operation. The plant configuration and component sizes used were those selected as the baseline design in the original study[4]. The control system design selected is as close as possible to that employed on conventional gas turbine ships with only those controls added which were necessary to accommodate the steam loop.

Simulations were developed from mathematical models of the major plant components, gas turbine/propeller controls, and several candidate control systems for the steam loop, and were exercised to select a baseline control system design. The detailed analysis that followed has allowed the questions posed at the beginning of this paper to be answered as follows:

1. What is the complexity and extent of the control system required for a COGAS propulsion plant?

As has been indicated, the baseline control system consists of separate gas turbine/propeller and steam plant control systems totally independent of each other. Steam plant performance and control after start-up is entirely "slaved" to the gas turbine exit conditions (mass flow and temperature) which are ultimately determined by engine throttle position. The steam turbine operates with no control of the steam to the steam turbine. Since the steam loop is designed to operate through the extreme ranges of steam turbine inlet pressure and temperature encountered with changing gas turbine power, the steam throttle can remain full open during normal operation.

The only additional controls required for this system are those typically provided for the feed and condensate system of any steam plant. These include: boiler drum, DFT and condenser level controllers, a DFT shell pressure controller, and condensate and feed pump controllers. All of these are fully automatic in response to steam plant conditions. Thus, there is little difference in the control devices or control methodology between the baseline COGAS plant and a conventional gas turbine plant from the operators point of view.

From a maintenance standpoint, there are additional local controllers. It should be noted, however, that of all the controls required on a conventional steam plant, those for the feed and condensate system are among the most reliable. For such a plant, many more maintenance hours are consumed on forced draft blowers, burners, automatic combustion controls and burner sequencing systems, none of which are required for a COGAS plant.

It must also be emphasized that since the gas turbine controls remain unchanged and independent from the steam loop controls, any casualties to the steam system will have no effect on the performance of the gas turbine.

2. Given the control system, what are the characteristics of a COGAS plant for acceleration, deceleration and steady state operation?

Maximum condition maneuvers including acceleration, crashback and crash-ahead were simulated with results indicating peak transmission and propeller loads typical of gas turbine plants. Gas turbine response to throttle commands is very rapid requiring only a few seconds to achieve large turbine exit changes (mass flow + temperature) during increases or decreases in power. Steam plant response time is much slower. The steam "time constant" (time required to achieve 63% change for full power excursions) is about 105 seconds. The excursion is not complete until about 300 seconds. These responses were obtained for metal and fluid masses typical of a lightweight, compact WHRU design such as that required for shipboard installation.

As might be expected, the dynamics of the gas turbine during crash maneuvers has little effect on the steam loop due to the wide dispairity in time constants. Quick changes in gas turbine power are simply not "seen" by the steam loop. The result, Figure 9, is a very stable system capable of power excursions of over 20,000 HP in seconds with an additional 7,000 HP which takes 2 to 5 minutes to respond.

These results were obtained for a gas generator speed governed gas turbine with no closed loop shaft speed governing. Because of the well behaved nature of this system, no additional governors or protective devices are required.

3. Are any special requirements associated with crashback or crash-ahead maneuvers?

One surprising result of this study was the lack of a need to dump steam during reversals. Due to the rather large inertia of the steam turbine and the characteristics of the propeller modelled, the propeller speed up at zero pitch was much less than anticipated. Because of this, an overspeed condition is not approached during crashback maneuvers, even though the steam turbine is producing nearly full power throughout the maneuver.

4. What are the values of peak thrust and transmission loads of a COGAS plant as compared to a simple cycle GT plant for crashback acceleration, and crashahead maneuvers?

With all other factors held the same, peak loads during crashback for a COGAS plant are somewhat higher than would be expected for a pure gas turbine plant of equivalent horsepower. This is due to the continuous power provided by the steam turbine during this maneuver.

It was demonstrated, however, that these high loads can be significantly reduced and at the same time ship performance improved, through optimization of the control system alone. In particular, the rate of pitch change can be slowed as it passes through zero pitch. Then, after the thrust peak has passed, 100% of installed power would be used. The result is a peak thrust limited to maximum ahead condition and a reduction in ship stopping time.

There are other control schemes which can be employed to reduce peak loading during reversals and acceleration which are being investigated as part of a separate study. The significant result from this preliminary work on control optimization is that a COGAS plant can be controlled to limit peak thrusts to steady state full ahead values without impairing ship maneuverability or performance.

5. Are there any special requirements associated with start-up and shut-down of a COGAS plant?

Start-up of a COGAS plant would be straightforward. The gas turbine would first be started, and circulation established in the steam loop with the steam turbine bypassed. After a steam head has been built up, the steam turbine throttle valve would be opened and the turbine would eventually pick up its share of the load. To prevent steam turbine windmilling losses while building up steam prior to opening the throttle valve, the steam turbine could be declutched or a vacuum taken on the condenser.

In a pure steam plant, start/stop procedures are normally accomplished locally in manual control. This is largely because of concern over boiler safety and the high complexity of most oil fired steam plants. The COGAS plant used in this study is simple by comparison, as the boilers are unfired. Thus, it becomes quite feasible, and desirable, to provide remote start-up and shut-down capability for the steam loop. This would be accomplished by a separate sequencing control similar to, but less complex than, that provided for the gas turbine.

6. What are the general conclusions with regard to the ease or difficulty in controlling a COGAS propulsion plant?

The plant and control system design studied, while not optimal, is among the least complex of the several possible COGAS configurations which have been proposed. Further refinements will undoubtedly be found desirable after the first prototype is built and tested as is the case for any new system. However, it has been shown that a COGAS plant can be designed with an uncomplicated control system capable of stable operation in the steady-state condition and throughout the full range of extreme ship maneuvers. No exotic control equipment is needed, no change in normal operating procedure is required, and peak loading can be kept within reasonable limits without sacrificing ship performance.

The economy and feasibility of the COGAS system have now been adequately demonstrated and the time has come to consider construction of a prototype plant.

ACKNOWLEDGEMENTS

The authors wish to acknowledge the computer simulation assistance in this study provided by T. R. Harper and L. C. Carroll of Propulsion Dynamics, Inc.

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APPENDIX A

NOMENCLATURE

N_E	Engine Speed
P_{ENG}	Engine Horsepower
PLA	Power Lever Angle (Gas Turbine)
PR	Propeller Pitch Ratio
W_F	Engine Fuel Flow Rate
ABYP	Normalized Effective Flow Area of Steam Dump Valve
A1S	Normalized Effective Flow Area of Main Steam Valve
A2S	Normalized Effective Flow Area of Extraction Flow Valve
CMIN	Minimum Heat Capacity of Gas Flow and Feedwater Flow
CP15G	Specific Heat, Flue Gas between Boiler and Economizer
CP16G	Specific Heat, Flue Gas between Superheater and Boiler
CP8	Specific Heat, Flue Gas at Gas Turbine Exit
EFE	Effectiveness, Economizer
EFF	Efficiency, Steam Turbine
EFSH	Effectiveness, Superheater
ETAST	Mechanical Efficiency, Steam Turbine
FSH	Flow Coefficient for Superheater
HD	Enthalpy, Feedwater Entering Downcomers
HDW	Enthalpy, Saturated Liquid in Steam Drum
HEXHS	Enthalpy of Exhaust of Ideal (isentropic) Steam Turbine
h_{fg}	Difference in Enthalpy of Saturated Fluid and Saturated Gas
HW	Enthalpy, Steam-Water Mixture Leaving Boiler
H14S	Enthalpy, Feedwater at Inlet to Economizer
H15S	Enthalpy, Feedwater Leaving Economizer

H16S	Enthalpy, saturated vapor leaving steam drum
H17S	Enthalpy, superheated steam leaving superheater
H3WS	Steam turbine exhaust enthalpy based on non-ideal expansion
KSH	Heat transfer coefficient for superheater
KVDW	Desired water volume in steam drum
KW8	Design flow at gas turbine exit
K1S	Steam turbine flow valve coefficient
MBLR	Mass of boiler
MECON	Mass of economizer
MSH	Mass of superheater
N_{ST}	Steam turbine speed
P_{ST}	Steam turbine power
P16S	Pressure, steam drum
P17S	Pressure, superheater outlet
QBLR	Heat transfer to water in boiler
QECON	Heat transfer to water in economizer
QSH	Heat transfer to steam in superheater
RDW	Density, saturated liquid in steam drum
R16S	Density, saturated vapor in steam drum
R17S	Density, steam at superheater outlet
SF	Entropy, saturated fluid at 3.0"Hg
SFG	Difference in entropy of saturated fluid and saturated gas at 3.0"Hg
S17S	Entropy, exit of WHRU
T14G	Temperature, flue gas at exit of WHRU
T15G	Temperature, flue gas between boiler and economizer

The
Society of Naval Architects and Marine Engineers

AN ADVANCED COMBINED CYCLE
MARINE POWER PLANT
FOR AN ARCTIC LNG TANKER

by

G.A. Clough

San Diego Section

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by

G.A. Clough

April 1980

For Presentation to the San Diego Section of
The Society of Naval Architects and Marine Engineers

Abstract

Combined cycle gas turbine propulsion utilizing waste heat recovery boilers are highly efficient plants. This paper discusses their application to an Arctic LNG tanker using LM2500 gas turbines and 9000 horsepower steam turbines for main propulsion.

Introduction

During the past ten years the price of fuel has increased by more than an order of magnitude. In addition, fuel availability has decreased to the point where fuel at any price is sometimes simply not available. Most projections for the future call for this situation to only get worse. All statements issued recently by the Organization of Petroleum Exporting Countries (OPEC) indicate that they will continue to raise prices and decrease output. This situation makes it necessary for shipowners to seek propulsion systems and operating policies which reduce the need for, and the expense of, fuel.

Prior to the Arab Oil Boycott, shipowners had been installing ever more powerful engines to get greater speeds. The Seatrain gas turbine powered container ships were an example of how speed could pay back high fuel use. However, with the unprecedented increase in fuel prices these ships were no longer economically attractive. They are now being refitted with diesel propulsion. The U.S. Navy made a commitment to gas turbine propulsion for their destroyer-type ships for the DD-963 and FFG-7 classes. The gas turbine system is ideally suited to the destroyer application where high power density and rapid response are required. However, the poor part-load performance of gas turbines is forcing the Navy to look at alternative systems.

With the OPEC countries adopting pricing and production policies which could be considered tantamount to blackmail, alternative sources of energy must be developed. Unfortunately, most of the easily produced fossil fuel has been discovered and is being used. Developing new sources of energy is often very difficult and often requires whole new technologies. Some of the more exciting finds in recent years have been in the Arctic regions of Canada and Alaska. One such find was made on Melville Island in the Canadian Arctic region. It has not yet been developed because there is no delivery system available for it.

Solar Turbines International has been developing a simple advanced technology combined cycle gas turbine and steam turbine system for its commercial market for several years and is currently participating in a U.S. Navy development program to develop a combined cycle system suitable for destroyer-type ship applications. This paper discusses how these systems could apply to a tanker to deliver LNG from the Arctic regions.

The Arctic Pilot Project

The Arctic Pilot Project is an ambitious project undertaken by Petro-Canada, The Alberta Gas Trunk Line Company and Melville Shipping Limited to develop a tanker to deliver liquified natural gas to markets. Natural gas has been discovered at Drake Point on Melville Island in the Canadian Northwest Territory. The reserve is estimated at 5.5 trillion cubic feet and is predicted to last the 20 years of this project. The gas will be extracted at Drake Point and sent by pipeline to a liquefaction plant on the southern coast of the island.

From the liquefaction plant, the LNG will be shipped to southern Canadian markets on a year-round basis. The ships used to do this will be large icebreaking tankers designed for a 140,000 cubic meter LNG capacity. The ships will be just over 1000 feet long and will be powered by three screws. The propulsion power planned is eight aircraft-derivative gas turbines, each driving a generator which will drive two electric motors per shaft. Cargo boil-off supplemented by vaporized cargo will be used as fuel.

As the name implies, this is a pilot project because it is small by comparison with other Arctic projects. It is intended to provide practical, proven operational experience for future projects to be based on. The reduced size of the project should reduce the impact of any problems encountered during the project.

The Combined Cycle

As a leading supplier of small and medium industrial gas turbines, Solar has been working for several years to find ways to offset the ever increasing fuel costs. This effort has resulted in the design and development of a simple combined cycle system for gas turbine waste heat recovery.

The combined cycle is the combination of two thermodynamic cycles into one very efficient power plant. The first thermodynamic cycle is the Brayton cycle used by the gas turbine. The second thermodynamic cycle is the Rankine steam cycle. These combined cycles are frequently referred to as COGAS (COmbined GAS turbine and Steam turbine). Figure 1 shows a combined cycle schematic.

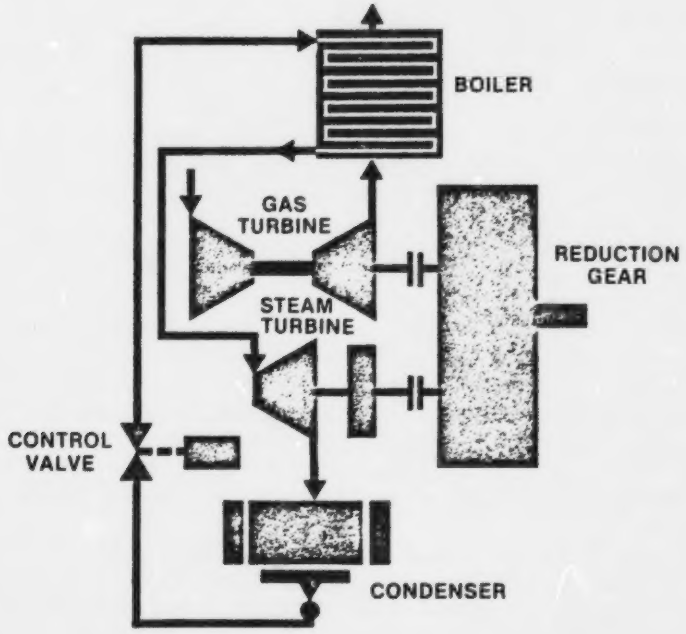

Figure 1. Combined Cycle Schematic

The combined cycle concept has received considerable attention in recent years because of its very attractive thermal efficiency. There are a number of combined cycles operating around the world, but few in shipboard applications, and these are open steam systems for heating, etc. There are a few combined cycles on diesel engines where the steam is used to drive the ship's service generators. This may be due to the perceived complexity of the system. It has been accused of being the worst of both worlds with the fuel sensitivity of the gas turbine and the problems and complexity of steam systems. In order for the combined cycle to be accepted, it had to be proven efficient, reliable and simple.

Solar undertook this project and in 1975 made a commitment to develop and produce a system with the desirable characteristics. Solar directed its efforts toward the commercial market which used the majority of their gas turbines. This direction demanded a plant that was efficient, simple, easily controlled, small size, modular, unmanned and that could be remotely operated.

At about the same time, the U.S. Navy was embarking on a project called RACER for RAnkine Cycle Energy Recovery. They have made a massive commitment to gas turbine propulsion for surface combatants and have funded the development of several gas turbines in the 1000 to 30,000 horsepower range. In 1975 the Navy was told to reassess its position on simple cycle gas turbines. The result was that the Navy conducted an extensive study on the best way to proceed and concluded that a combined cycle warranted immediate development.

The Navy combined cycle development program (RACER) was specified to use the LM2500 gas turbine exhaust to provide heat for a steam system to produce additional power. The design goals were simply stated as: high reliability, low maintenance, five year overhaul cycle, low size and weight, and the best performance that the preceeding goals could give.

The Navy program is currently only about half way into the first phase of a two phase program. Solar's commercial program is much further along with a prototype plant constructed and operating at this time. Although the power levels are somewhat different, the concept is the same and easily scaled to power. The results to date have been most rewarding. The predicted thermal efficiency, using Solar's Centaur gas turbine, was 37 percent. The actual thermal efficiency achieved is 40 percent. The concept is proving to be very practical. A brief description of a system appropriate for use with the LM2500 follows.

Steam System

Because the objective of this system is to increase efficiency, and because the steam system is slaved to the gas turbine exhaust conditions, the steam pressure is allowed to float. This provides for optimized steam cycle conditions over a wide operating range. Many computer analyses were made to determine the optimum pressure which resulted in the boiler being designed for about 400 psig at approximately half power and over 700 psi at full power. In addition to providing the most efficient steam cycle, it also makes the system control simpler.

A second important design decision was to utilize passive feedwater treatment. The desirability of this is readily apparent when complexity and manning are considered. To do this requires that the feedwater makeup requirement be kept extremely low. The system goal is less then 20 gallons per day. To do this requires that the system have as few components as possible and that these components be virtually leak-free. This goal has been obtained and will be explained later.

The Components

Steam Generator

The heart of the proposed cycle is the steam generator. The requirements for steam generator control can easily complicate the system and the need for large volumes of water can raise its response time to unacceptable levels. The steam generator used is a very simple once-through steam generator (Figure 2) with no drums. Each circuit is a complete steam generator in one tube. Water is fed into the inlet, heated, vaporized and superheated. Each circuit consists of many rows, with size determined by power requirements. The steam generator is unfired and looks much like a conventional economizer when viewed from the gas path side.

The tubes are stainless steel and finned for more efficient heat recovery. The boiler is constructed to ASME code standards. It can be run dry without damage which is, in fact, the method used to clean the gas side.

The steam generator can be either dual or single pressure as desired. Because of the control method, the added complexity due to dual pressure is minimal. Additional heat may be recovered by selecting a dual pressure system because of pinch point considerations. Figures 3 and 4 show the additional heat recovery possible with dual pressure.

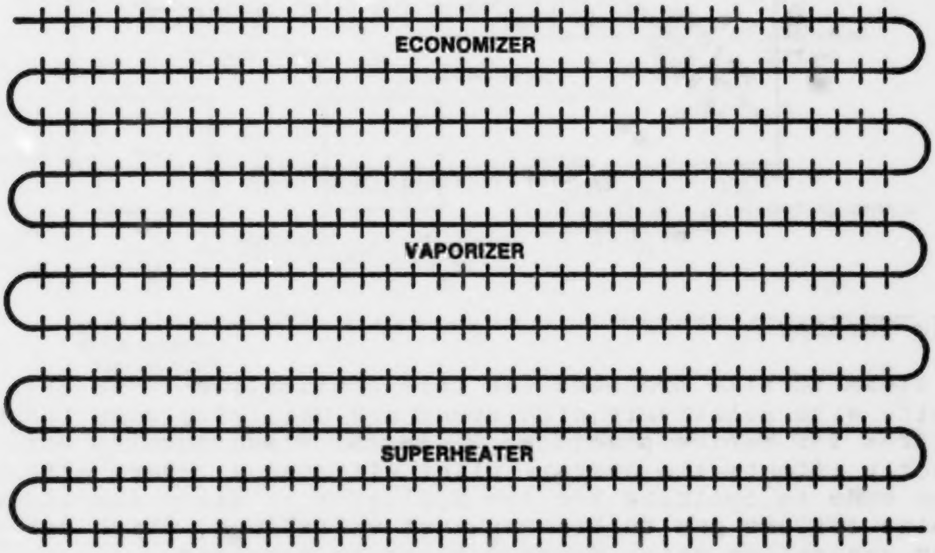

Figure 2. Once-Through Steam Generator Schematic

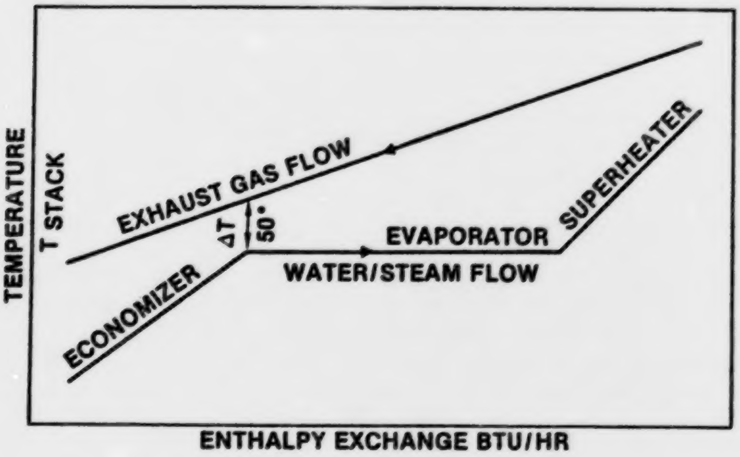
TEMPERATURE
T STACK
EXHAUST GAS FLOW
ΔT
50°
EVAPORATOR
SUPERHEATER
WATER/STEAM FLOW
ECONOMIZER
ENTHALPY EXCHANGE BTU/HR

Figure 3. Single Pressure Cycle

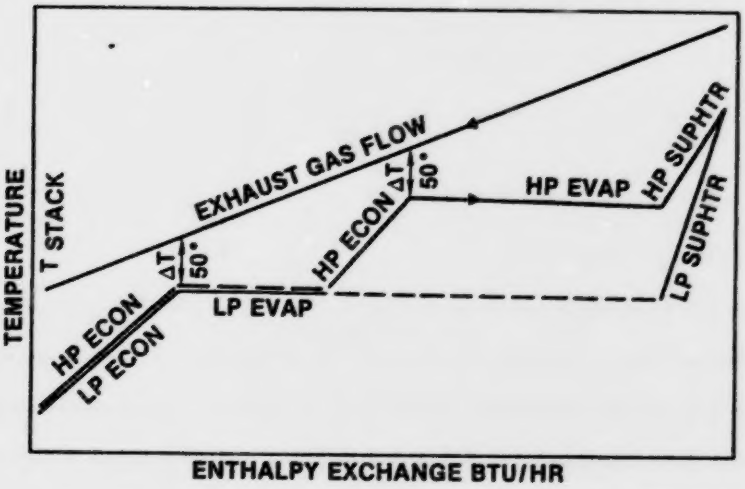
TEMPERATURE
T STACK
EXHAUST GAS FLOW
ΔT
50°
HP EVAP
HP SUPHTR
HP ECON
ΔT
50°
LP EVAP
LP SUPHTR
HP ECON
LP ECON
ENTHALPY EXCHANGE BTU/HR

Figure 4. Dual Pressure Cycle

Steam Turbine

The steam turbine proposed for this application is an axial flow turbine with relatively high speed and utilizing some innovative features for marine practice. Since the steam turbine efficiency directly affects the overall plant efficiency, every effort has to be made to optimize for the application. Since the steam turbine follows gas turbine exhaust variations, which determine steam generation rate, no throttling is required.

Water Treatment

Water purity is critical to the unmanned and low maintenance requirements. The first step in the water treatment is to keep leakage as low as possible. A condensate demineralizer is included to remove dissolved solids. The demineralizer is sized to provide a long service life between recharges. A filter is added downstream of the demineralizer to separate out any solid particles which may carry over into the system.

Plant Control

The steam plant is controlled with one valve under operating conditions. The steam cycle is slaved to the gas turbine with steam pressure varying as gas turbine power varies. A solid state microprocessor is utilized (Figure 5). With the once-through boiler, the output steam pressure and temperature is directly related to the feedwater flow rate. The microprocessor senses the output steam temperature of the boiler and the gas turbine exhaust gas temperature and controls the feedwater flow to maintain the desired temperature control.

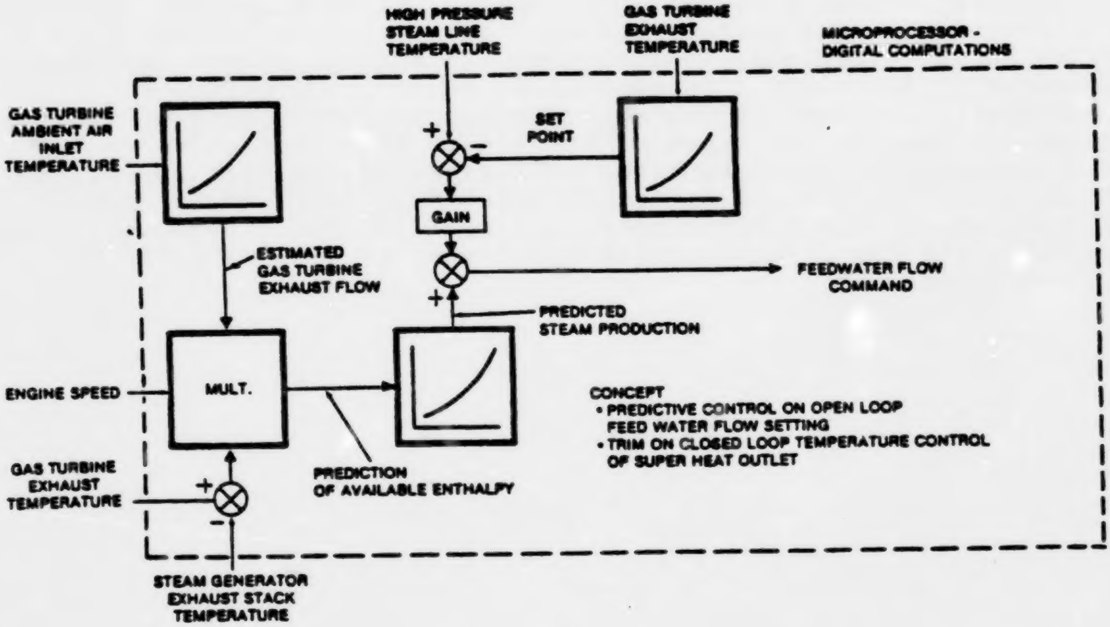
HIGH PRESSURE STEAM LINE TEMPERATURE
GAS TURBINE EXHAUST TEMPERATURE
MICROPROCESSOR - DIGITAL COMPUTATIONS
GAS TURBINE AMBIENT AIR INLET TEMPERATURE
+
−
SET POINT
GAIN
ESTIMATED GAS TURBINE EXHAUST FLOW
FEEDWATER FLOW COMMAND
+
PREDICTED STEAM PRODUCTION
ENGINE SPEED
MULT.
CONCEPT
• PREDICTIVE CONTROL ON OPEN LOOP FEED WATER FLOW SETTING
• TRIM ON CLOSED LOOP TEMPERATURE CONTROL OF SUPER HEAT OUTLET
GAS TURBINE EXHAUST TEMPERATURE
+
−
PREDICTION OF AVAILABLE ENTHALPY
STEAM GENERATOR EXHAUST STACK TEMPERATURE

Figure 5. Combined Cycle Feedwater Control

Operating Profile

The Arctic Icebreaking LNG Tanker has a very high power requirement while transiting ice fields. The most arduous duty is in the ice ramming cycle.

The ice ramming cycle starts with the ship running at full power in ice which is heavy enough to reduce the momentum of the ship. As the ice reduces the ship speed to zero, the power is reduced to a minimum, reversed and increased back to maximum. When the ship has backed an appropriate distance, power is once again reduced to a minimum and direction reversed to the forward direction. The power level is again brought to full power to ram the ice. It is estimated that the ship would conduct about 1000 of these cycles each year.

Because the steam cycle follows the gas turbine and is slaved to it, this cycle is much more able to respond to these types of transients than a conventional boiler. There is a very low water inventory in the system generator which reduces thermal lags to an absolute minimum. In addition, the control system includes a throttle valve and a bypass valve. While these valves are normally utilized in the start-up and shut-down mode only, they may be used in the types of transients described above in order to ensure the rapid response rate required. In this cycle, the throttle valve would close and the bypass valve would open as soon as the gas turbine power was reduced. The bypass valve would close and the throttle valve would open as soon as power was reapplied.

Power Plant Performance

For this application the use of LM2500 gas turbines was assumed. The system performance is shown in Table 1. The specific fuel consumption (sfc) of the system at 59°F, using fuel with a lower heating value of 18,400 Btu/lbm, is 0.290 lb/hp-hr. This is equivalent to an overall thermal efficiency of 47.8 percent. On liquified natural gas the heat rate is 5166 Btu/hp-hr for a thermal efficiency of 49.2 percent.

Table 1. Combined Cycle Performance

	NATURAL GAS	LIQUID FUEL
Simple Cycle Output (hp)	26,400	24,000
Heat Rate (Btu/hp-hr)	6,927	7,323
Combined Cycle Output (hp)	35,400	33,000
Combined Heat Rate (Btu/hp-hr)	5,166	5,326
Combined Thermal Efficiency (%)	49.2	0.290
SFC (lb/shp-hr)	-	0.290

Summary

The power plant described in this paper is similar to that being developed for the U.S. Navy. A commercial power plant at lower power levels has been developed and is now being operated at Solar.

The Arctic Icebreaking LNG Tanker is an ideal application of a combined cycle power plant. The ready availability of natural gas fuel makes the use of gas turbines very attractive. Turbines can be designed to run on either natural gas or diesel fuel as a back-up. Gas turbines have a very high power density and a very rapid power response capability. The steam cycle slaved to the gas turbine can respond quite rapidly and can be well within required limits when using electric drive. This advanced steam system is considerably lighter and more compact than conventional steam systems. Ship designers who desire highly efficient propulsion plants should consider the benefits that are offered by the combined cycle system.

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stag

Combined-Cycle
Power Systems

STAG COMBINED-CYCLE OPERATING EXPERIENCE

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Manager—STAG Sales & Projects
Medium Steam Turbine Department

GENERAL ELECTRIC

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INTRODUCTION

The modern STAG* combined-cycle plant used for electric utility power generation has evolved from the earliest gas turbine applications to its present mature form through a variety of applications. This history has established a series of design parameters which result in excellent efficiency and operating flexibility while maintaining lower overall costs than conventional generating plants. It is important to look at the base of experience before discussing the modern plants in order to better understand these superior characteristics.

BACKGROUND

During the 1960's General Electric built over 100 varieties of combined-cycle plants for industrial and utility applications. These ranged from base-load process steam plants to utility repowering applications where existing steam plants were operated daily using a new gas turbine and heat recovery steam generator to replace worn-out power boilers. During the 1960's four small municipal utilities purchased single-shaft STAG plants. In this application, the generator is driven on one end by the gas turbine and the other end of the shaft is directly connected to the steam turbine, all on single-shaft. The City of Ottawa (Kansas) and the City of Hutchinson (Minnesota) purchased units utilizing the MS3000 gas turbine while the Wolverine Electric Cooperative (Big Rapids, Michigan) and the City of Clarksdale (Mississippi) plants utilized the larger MS5000 gas turbine. The ratings of these plants were dependent on the MS3000 and MS5000 gas turbines which ranged from 10-25 MW each, with the MS5000 single-shaft version having a capability of about 35 MW. The performance in service of these units was excellent. For example, the City of Clarksdale machine is still being used as the city's base load power generation prime plant and operates every day for this municipality. This excellent experience led to the expansion of combined-cycle applications in the USA. These later and larger plants will be the subject of this paper.

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*Trademark of the General Electric Company.

STAG-1-77A

stag

Combined-Cycle
Power Systems

STAG COMBINED-CYCLE OPERATING EXPERIENCE

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Medium Steam Turbine Department

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STAG COMBINED-CYCLE MATURATION

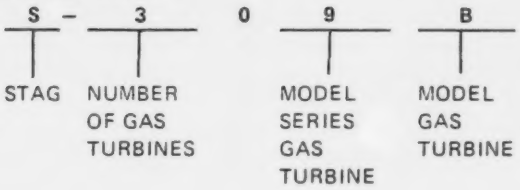
Beginning in the 1970's several major utilities decided to build larger plants using multiple gas turbines with heat recovery steam generators to drive larger single steam turbines. The status of these larger generation STAG plants as of August 1978 is summarized below:

- 20 STAG Plant Sites
- 34 STAG Plants in Service and On Order
- 23 In Commercial Operation
- 2 In Installation
- 9 in Engineering

Plant Designations — In this paper you will note that General Electric designates its plants by a letter and number sequence. This is shown in the figure below.

PLANT DESIGNATION

EXAMPLE: S309B



The first large multi-shaft STAG system was purchased by Jersey Central Power and Light (New Jersey) in 1971, but the dawn of the new era of STAG systems was in the summer of 1972 when, during a three-month period, four utilities ordered seven STAG plants; four multi-shaft plants and three single-shaft 107 plants. Tables I and III list, in approximate start-up sequence, the 34 STAG plants shipped or on order. Note that they are about equally divided between multi-shaft and single-shaft STAG arrangements.

Some of the variety of arrangements and plant configurations as installed are shown in site photos for Houston (Texas) Lighting & Power (Fig. 1), Portland (Oregon) General Electric (Fig. 2), Duquesne (Pennsylvania) Light Company (Fig. 3), Salt River Project (Arizona) (Fig. 4), Arizona Public Service (Fig. 5) and Puerto Rico Water Resources Authority (Fig. 6). We have applied STAG 107, 207, 307, 405, 407, 507, and 607 systems, and they have employed MS5000 gas turbines as well as MS7000B, MS7000C and MS7000E units. Both fired and unfired HRSGs have been used in the STAG plants and the steam turbines have ranged from single-flow units with 17 inch (43 cm.) last-stage buckets to four-flow units. This experience is an extremely valuable base for future General Electric STAG plant designs.

TABLE I

GENERAL ELECTRIC STAG EXPERIENCE (TURBINES)

UTILITY	STATION	PLANTS	GAS TURBINE	STEAM TURBINE
Houston Lighting & Power Co.	TH Wharton	(2) S-407	(8) MS7001B	(2) DF-23
Duquesne Light Co.	Brunot Island	(1) S-307F	(3) MS7001B	(1) DF-23
Salt River Project	Santan	(4) S-107	(4) MS7001B	(4) SF-17
Ohio Edison Co.	West Lorain	(1) S-207F	(2) MS7001B	(1) DF-20
Arizona Public Service	West Phoenix	(3) S-107	(3) MS7001C	(3) SF-17
Iowa-Illinois Gas & Electric Co.	Moline	(1) S-405	(4) MS5001LA	(1) SF-20
Puerto Rico Water Resources Authority	Aguirre	(2) S-407	(8) MS7001B	(2) DF-20
Jersey Central Power & Light Co.	Gilbert	(1) S-407F	(4) MS7001B	(1) DF-23
Western Farmers Electric Cooperative	Anadarko	(3) S-107	(3) MS7001E	(3) SF-17
Portland General Electric Co.	Beaver	(1) S-607	(6) MS7001B	(1) 4F-16.5
Korea Electric Co.	Kunsan, Yongwol	(2) S-407	(8) MS7001B	(2) DF-23
Pacific Gas and Electric	Bay Area	(1) S-407	(4) MS7001E	(1) DF-26
Mass. Municipal Wholesale Electric Co.	Stony Brook	(1) S-307	(3) MS7001E	(1) DF-23
Southern California Edison Co.	Lucerne Valley	(3) S-507	(15) MS7001E	(3) 4F-16.5
Pacific Gas & Electric	Thermal A & B	(4) S-407	(16) MS7001E	(4) DF-26

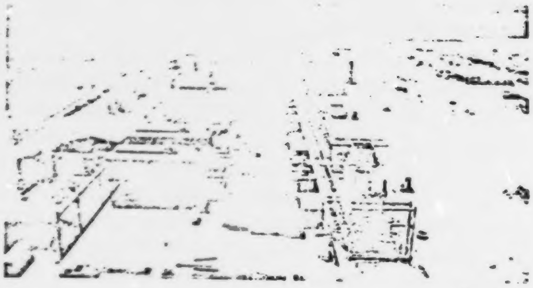


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Figure 1

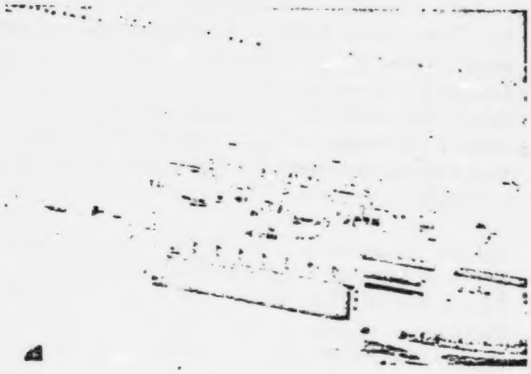

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Figure 4

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Figure 2

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Figure 5

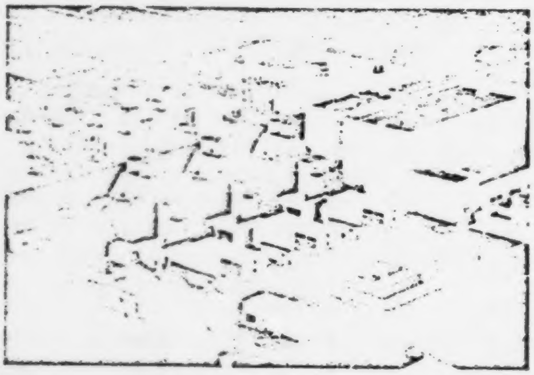

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Figure 3

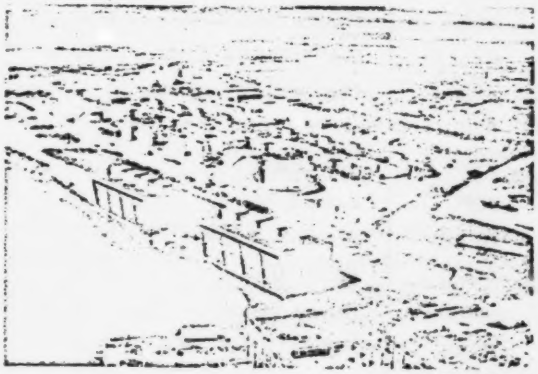

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Figure 6

Table II continues the description of the General Electric STAG plants. Seven architect engineers have participated in these plants, including General Electric's Project Engineering Operation (PEO), while Southern California Edison and Pacific Gas & Electric have elected to provide their own plant engineering services. The STAG plants have been designed for most fuels with a wide variety of steam conditions.

Plant ratings range from 72 MW for the single-shaft Salt River STAG 100 to 545 MW for the Portland (Oregon) General Electric STAG 607 system. STAG plant efficiency continues to improve, as illustrated by the Pacific Gas & Electric STAG 407 system which has a heat rate of 7930 Btu/kWh (1998 Kcal). This STAG employs the deaerating steam supplied feedwater heating (DASSH) cycle. The Massachusetts Municipal Wholesale Electric Company (MMWEC) STAG 307 system continued the improvement as well by using a DASSH steam cycle with admission at turbine cross over. The heat rate guarantees for General Electric STAG plants range up to 10,000 Btu/kWh (2523 Kcal) for the Iowa-Illinois Gas & Electric STAG 405 plant using the two heater cycle. The STAG plants employing the deaerating condenser cycle generally have heat rates falling between those with DASSH cycle and those with the two heater cycle, but primary fuel also influences guaranteed heat rate.

Part of the General Electric STAG plant experience success story is revealed in Table III. In spite of the large influx of orders for STAG combined-cycle systems during the summer of 1972, shipments were completed in less than one and one half years, with steam turbine synchronization occurring within two years of order date. It takes a well coordinated team of factory, architect/engineer, installation contractor and utility people to achieve the enviable record of commercial operation in two to two and one half years from order date.

As an example of the General Electric commitment to meeting customer needs, shipments for the Korea Electric Company STAG 407 units started about six months from date of order, completing shipping in 17 months. The gas turbines operated in the summer of 1977 while the steam plant is being completed.

TABLE II

GENERAL ELECTRIC STAG EXPERIENCE (RATINGS)

			STEAM CONDITIONS		PLANT RATING (SITE CONDITIONS, DISTILLATE FUEL)		
UTILITY	ARCHITECT/ENGINEER	FUEL	PRESS. ATO	°C	MW	HR KCAL/KWH	CYCLE
HL&P	Ebasco Services	G/D	56.3	451	297	2194	DC
DLC	Gibbs & Hill	D	87.9	509	326	2182	2 HTR
SRP	General Electric	D	33.0	454	72	2172	DC
OE	Commonwealth Assoc.	D	87.9	509	222	2182	2 HTR
APS	Commonwealth Assoc.	D/R	33.7	426	83	2177	DC
IIGE	Stanley Consultants	D	59.8	482	95	2523	2 HTR
PRWRA	General Electric	D/R	64.7	460	292	2215	2 HTR
JCPL	Burns & Roe	D	87.9	509	319	2278	2 HTR
WFEC	Sanderson & Porter	G/D	42.2	460	91	2119	DC
PGE	Ebasco Services	D	64.7	460	545	2151	2 HTR
KECO	General Electric	D	59.8	460	318	2121	DASSH
PG&E	PG&E	D	59.8	509	406	2063	DASSH
MMWEC	Bechtel	D	59.8	509	346	1998	DASSH
SCE	SCE	D/G	59.8	468	431	2106	DASSH

TABLE III

GENERAL ELECTRIC STAG EXPERIENCE (INSTALLATION CYCLES)

UTILITY	UNIT	ORDER	SHIP CYCLE COMPLETE (MONTHS)	PLANT START-UP	INSTALL. TIME (MONTHS)	COMM. OPERATION	ORDER TO COMM. (MO.)
HLP	Wharton 3	5/72	15	5/74	9	8/74	27
	4	5/72	17	7/74	9	8/74	27
DLC	Brunot Is.	5/72	16	5/74	8	6/74	25
SRP	Santan 1	8/72	15	8/74	9	10/74	26
	2	8/72	15	12/74	13	12/74	28
	3	8/72	16	10/74	10	10/74	26
	4	5/73	16	4/75	7	5/75	24
OE	West Lorain	7/72	16	12/74	13	12/74	29
APS	W.P. 1	6/73	15	6/76	21	6/76	36
	2	6/73	16	5/76	19	6/76	36
	3	6/73	17	5/76	18	6/76	36
IIGE	Moline	2/74	22	12/76	12	5/77	39
PRWRA	Aguirre 1	12/73	20	4/77	20	6/77	42
	2	12/73	24	12/76	12	5/77	41
JCPL	Gilbert	8/71	22	3/77	45	5/77	69
WFEC	Anadarko 4	12/74	19	9/77	14	9/77	33
	5	12/74	20	10/77	14	10/77	34
	6	12/74	21	11/77	14	11/77	35
PGE	Beaver	6/74	29	9/77	10	10/77	40
KECO	Kunsan, Yongwol	7/76	IN ENGINEERING OR INSTALLATION PHASE				
PG&E	Bay Area	5/76					
MMWEC	Stony Brook	8/77					
SCE	L.V. 1	7/75					
	2	7/75					
	3	7/75					

The Southern California Edison contract, at 1,300 MW, represents one of the world's largest combined-cycle plants. The order of these three STAG plants was placed in July, 1975 but only the advanced engineering portion has been released. Southern California Edison's power generation plan has listed the Lucerne Valley plant as being able to meet various power requirements starting in 1981, with completion in the summer of 1985.

Even surpassing the SCE contract are two orders from Pacific Gas and Electric for five STAG 407E's for a total output of 2075 MW. One of these STAG 407's plants is for the Bay Area. The other four units will be installed at Pittsburg, California with combined cycle output of 1660 MW and commercial operation planned for the 1983 through 1985 time period. These PG&E plants will represent the worlds most efficient thermal power plants.

Table IV lists operating data on the STAG plants in service prior to 1978 and lends perspective to operation of the units. The total operating hours of 377,720 provides significant proof that these plants have reached maturity and can be counted on for critical generation.

Data on individual plant availability, reliability, efficiency and capability is difficult to obtain on a consistent

TABLE IV

UTILITY STAG EXPERIENCE
OPERATION HOURS

UTILITY	STARTED COMMERCIAL OPERATION	OPERATION HOURS
City of Clarksdale	6/72	44,649
City of Hutchinson	7/72	41,942
Wolverine Elec. Coop	6/68	71,847
Ottawa L&P	6/68	77,815
Iowa Illinois	4/77	3,378
Salt River Project 4-S 107B	10/74	48,714
Houston L&P Unit 3	12/74	19,605
Unit 4	12/74	17,932
Duquesne Light Co.	6/74	10,931
Ohio Edison	12/74	11,052
Arizona Public Service	6/76	12,329
Jersey Central	5/77	942
Puerto Rico Unit 1	6/77	3,777
Unit 2	5/77	2,541
Western Farmers Electric Coop	9/77	10,098
Portland General Electric	10/77	68
Total		377,730

As of July 1, 1978

basis and even more difficult to interpret. However, the available data on several of the plants indicates availability higher than on comparable large fossil or nuclear plants, in excess of 80 percent. This record is achieved even with the severe cyclic duty imposed on the mid-range STAG plants.

Recently we asked a number of Utilities who had purchased STAG plants in the early 1970's, how the STAG plants were being used and what advantage they found in having the STAG combined-cycle plants as part of their utility system. The answers that we received somewhat reflect the versatility of the combined-cycle plant. The following is a sample of the comments we received from this recent survey:

"Our STAG plants are used for seasonal base load operation. They are also used for system voltage regulation because of their fast response. The short installation time and low installation cost are very attractive."

"Our STAG plants have the lowest total power generation cost in the system."

Another utility responded: "The STAG plants are used for seasonal base load operation. They are operated twenty-four hours per day, seven days per week with excellent availability. They are the most efficient units on the system with production cost 80 percent of that for fossil steam plants on our system."

A Western USA Utility wrote: "The STAG plant is utilized for standby power only and peaking duty as we are basically a hydro and nuclear base load utility. The biggest advantage we see for the STAG plant is its rapid start capability and excellent heat rate. It also puts us in a favorable bargaining position for purchase power due to its relatively low cost of generation compared to the simple-cycle gas turbine operations."

A mid-West utility answered that the STAG plant has been utilized as a mid-range plant operating 12 to 16 hours per day, five days a week and is a vital part of their system. The advantage the STAG gives is the ability to generate fast, efficient power during their peak load periods. It also provides better availability than many of their coal fired plants. The STAG plant carries the best heat rate on their system.

Finally, a large utility stated the STAG plant is used as a mid-range plant with daily starting and stopping. During the off seasons and due to its high efficiency and fast starting capability, the STAG plant is used as the highest priority reserve for planned outage and emergency duty. High starting and operating reliability are extremely important. In the off peaks in the fall and spring, the combined cycle is used to allow planned maintenance for other plants and to offset losses of other plants due to forced outages. This plant is operated approximately 3500 hours a year.

SUMMARY

The first generation of small STAG plants, purchased in the 1960's, provided a successful experience base and the confidence to purchase the succeeding generation of larger STAG plants. As of mid-1978, the current generation of operating STAG plants, purchased in the early 1970's, is providing a wider experience base and proof of the economics of STAG systems for future mid-range and base load operation. Experience has now established that no other proven form of generation can achieve the efficiency of STAG plants, and at a capital cost far below that of conventional fossil and nuclear plants.

The
Society of Naval Architects and Marine Engineers

A PROPOSED COMBINED CYCLE MARINE POWER PLANT

by

N. L. MacIntyre

San Diego Section

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MARINE POWER PLANT

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January 1979

Abstract

Since 1975 Solar Turbines International has been developing a sophisticated combined cycle steam design which, when used in conjunction with Solar's new Mars gas turbine, can provide the military or commercial ship operator with a highly efficient yet straight-forward power plant. The specific fuel consumption of this engine system is competitive with the most advanced diesel engines.

The system design philosophy followed by Solar will be discussed and the major system components described. Installation and operational considerations will be presented and discussed.

System performance will be discussed briefly as it relates to economy of ship operation. Control and monitoring philosophy will be presented.

Introduction

Continuously rising fuel costs have been a major consideration for ship operators in recent years and this trend shows no likelihood of slowing down. Owners and operators considering new construction must carefully examine all practical approaches to maximize overall propulsive efficiency and thereby hold fuel usage to the lowest possible levels.

This powerful economic influence has contributed to the resurgence of interest among U.S. flag owners in high horsepower diesel engines in the last few years. Major suppliers of conventional steam systems for ship propulsion recognize this trend and have oriented their sales and marketing efforts to contain the sometimes labeled "Diesel Threat". Diesel engine proponents have been somewhat discouraged by the reluctance of American owners to rapidly shift to diesel even in view of the potential of significant fuel savings. This may be due to a fear that diesels are more "finicky" than steam systems and that unacceptably high maintenance costs might result.

The desire among ship owners and operators to have propulsion systems with favorable fuel rates yet low shipboard maintenance requirements is one of the factors that has influenced Solar Turbines International to develop a simple, but very efficient combined gas turbine (Brayton) and steam (Rankine) cycle power plant. The system described here is based on Solar's 10,000 HP class Mars gas turbine and will produce approximately 14,000 to 15,000 shaft horsepower in marine applications.

Basic system description

As commonly used, the term "Combined Cycle" refers to an engine system based on a simple cycle gas turbine whose exhaust heat is recovered in a heat recovery boiler to power a closed steam turbine system. The basic schematic of the system is shown in Fig. 1. This concept has been used with great success for many years, particularly in large sizes (e.g., 400 MW) by electric power utility companies for base-load power plants with advantageous fuel rates.

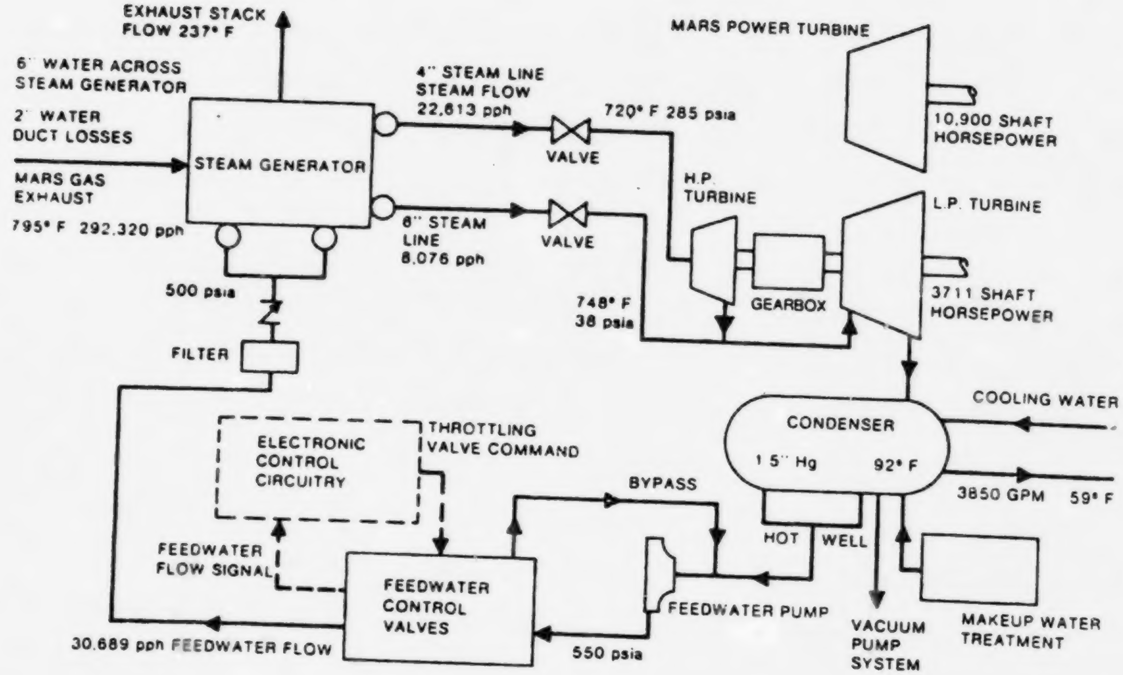
A design team was established to identify the system and component performance and characteristics necessary to develop this concept for lower horsepower applications. Accordingly, the following basic design goals were established:

- High efficiency
- Simplicity
- Complete automation
- Minimal package size
- Modular assembly

In order to meet these goals the design team was tasked with reviewing the basic concepts of each component and cycle design.

Cycle selection

The design goal of high efficiency dictated a review of the basic steam cycle and an analysis of possible cycle combinations and steam conditions, from a purely thermodynamic point of view. This evaluation identified the dual pressure steam cycle as being clearly superior to the more commonly used single pressure cycle. This is principally due to the fact that the Rankine cycle working fluid (steam/water) more closely follows the gas turbine exhaust temperature through the boiler with the dual


EXHAUST STACK FLOW 237° F
6" WATER ACROSS STEAM GENERATOR
2" WATER DUCT LOSSES
MARS GAS EXHAUST
795° F 292,320 pph
STEAM GENERATOR
500 psia
FILTER
4" STEAM LINE STEAM FLOW 22,613 pph
VALVE
720° F 285 psia
6" STEAM LINE 8,076 pph
VALVE
748° F 38 psia
MARS POWER TURBINE
10,900 SHAFT HORSEPOWER
H.P. TURBINE
GEARBOX
L.P. TURBINE
3711 SHAFT HORSEPOWER
CONDENSER
1.5" Hg
92° F
COOLING WATER
3850 GPM
59° F
HOT WELL
VACUUM PUMP SYSTEM
MAKEUP WATER TREATMENT
ELECTRONIC CONTROL CIRCUITRY
THROTTLING VALVE COMMAND
BYPASS
FEEDWATER FLOW SIGNAL
FEEDWATER CONTROL VALVES
FEEDWATER PUMP
30,689 pph FEEDWATER FLOW
550 psia

NOTES:

1. Auxiliary power is included (3%).
2. The data shown is for the power plant sizing case with the gas turbine at ISO conditions with steam generator losses included.

Fig. 1 Mars combined cycle

pressure system. This is illustrated in Fig. 2. In practical terms the dual pressure cycle results in a 12% gain in power output over a single pressure cycle, using the same design constraints in both cases. Final steam conditions were established as follows:

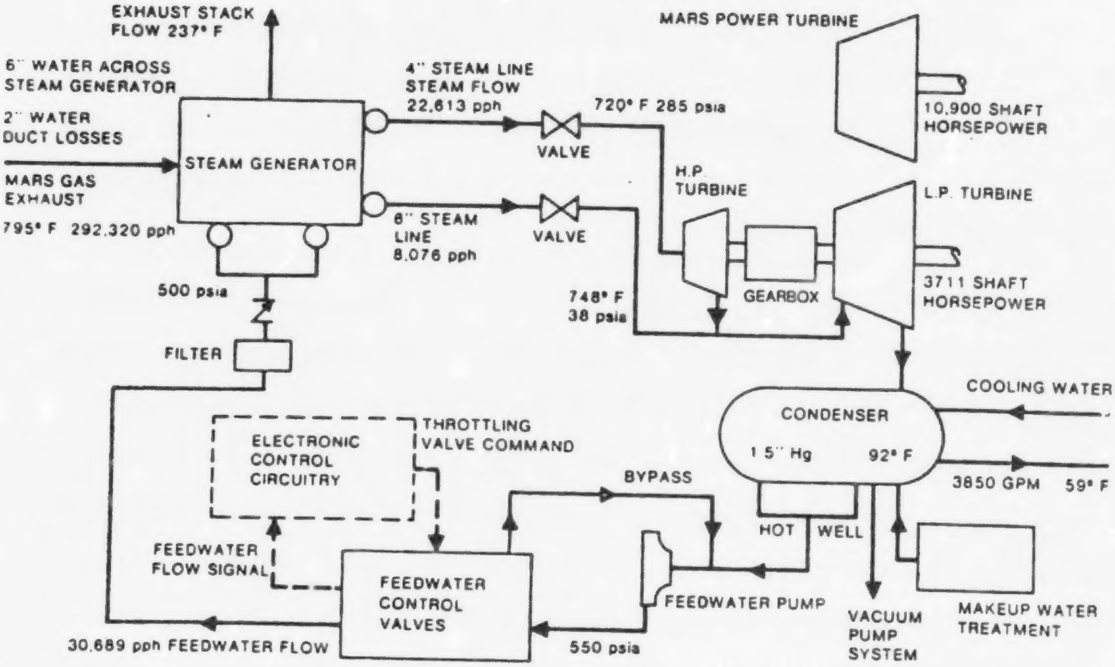
	Pressure (psia)	Temperature (°F)
High Pressure	285	720
Low Pressure	38	748

Boiler design concept

At first glance, the design goal of overall system simplicity appears to be adversely affected by the selection of a dual pressure steam cycle. Further, typical single pressure shipboard boilers present complex automation problems even when only feedwater control is considered.

Again, the design team identified, evaluated and selected a design approach which could satisfy the original goals. The "once-through" boiler, when properly designed, offers simple control of steam generation and feed rate and can be adapted for dual pressure use with little added complexity. A schematic of the once-through dual pressure boiler is shown in Fig. 3. This boiler has no drums and therefore no need for water level control. Further, steam generation rate is controlled by varying feed water rate directly without the need for multiple control elements as is common with drum-type boilers.

The boiler developed uses finned tubes for efficient heat recovery. Special stainless steel material was selected to prevent waterside corrosion. This material selection permits operation without conventional feedwater treatment and its attendant complications. Closed loop corrosion testing and full scale boiler module operations have confirmed this concept. (Deionization of makeup feedwater is used to prevent waterside deposit buildup.) Fig. 4 shows the test boiler module used to confirm design concepts as development has progressed.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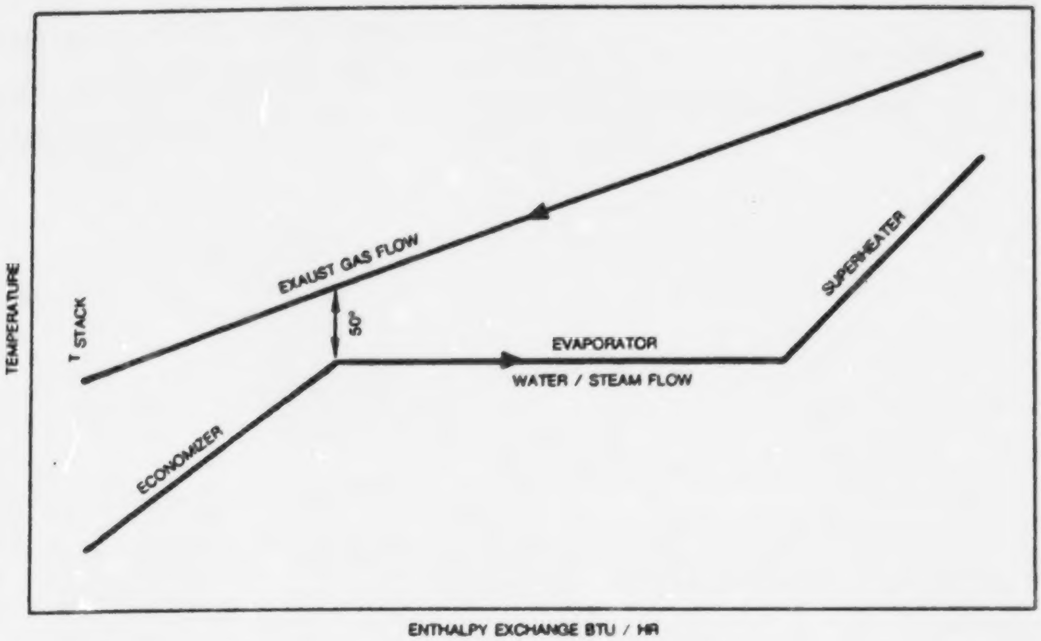
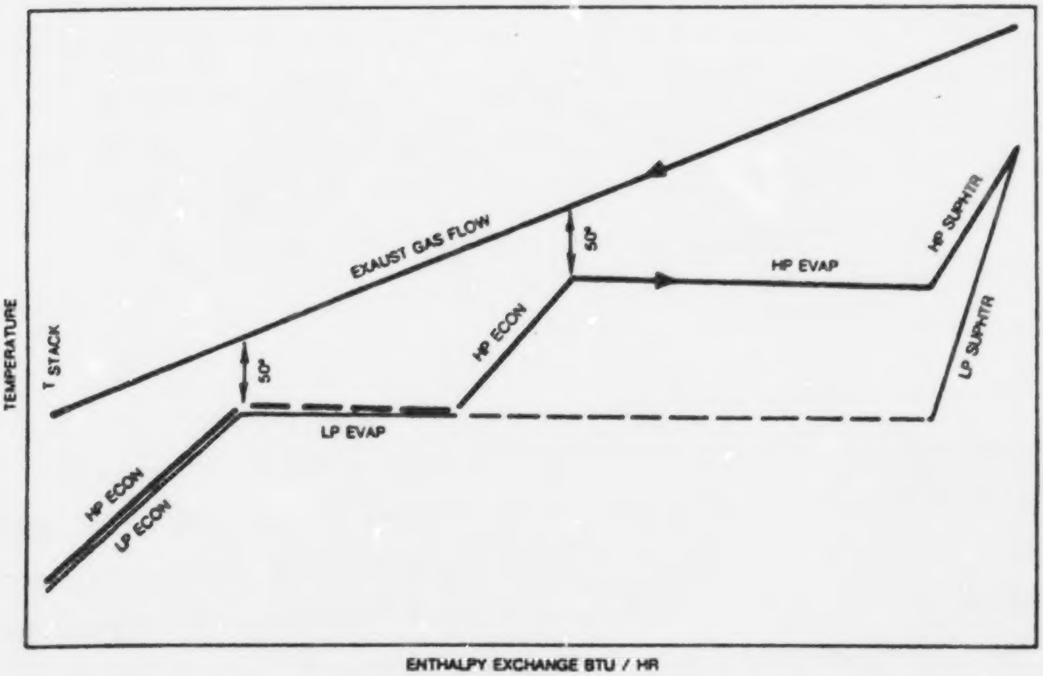

TEMPERATURE
T STACK
EXAUST GAS FLOW
50°
EVAPORATOR
WATER / STEAM FLOW
SUPERHEATER
ECONOMIZER
ENTHALPY EXCHANGE BTU / HR

a. Single pressure cycle


TEMPERATURE
T STACK
EXAUST GAS FLOW
50°
50°
HP EVAP
HP SUPHTR
LP SUPHTR
HP ECON
LP EVAP
HP ECON
LP ECON
ENTHALPY EXCHANGE BTU / HR

b. Dual pressure cycle

Fig. 2 Pinch point diagrams

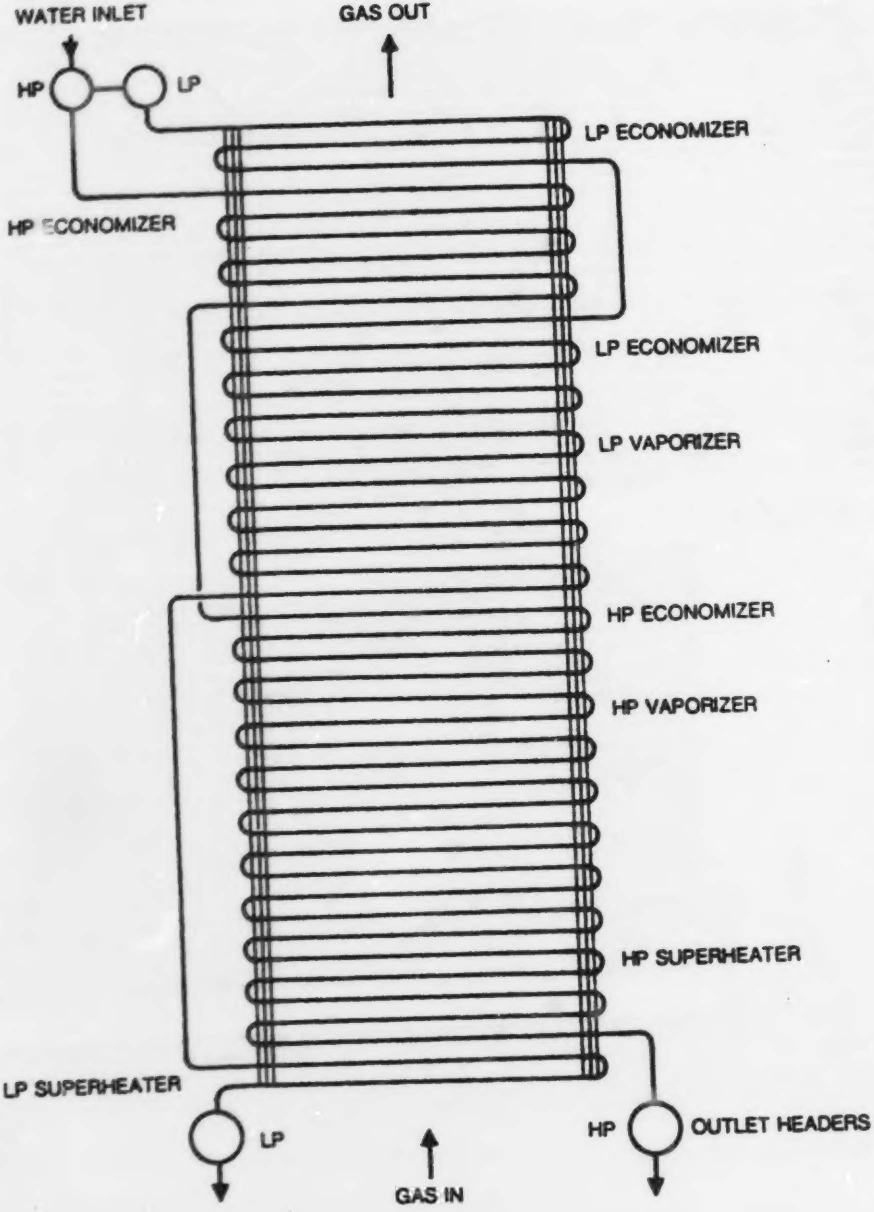

WATER INLET
GAS OUT
HP
LP
LP ECONOMIZER
HP ECONOMIZER
LP ECONOMIZER
LP VAPORIZER
HP ECONOMIZER
HP VAPORIZER
HP SUPERHEATER
LP SUPERHEATER
LP
GAS IN
HP
OUTLET HEADERS

Fig. 3 Mars combined cycle boiler

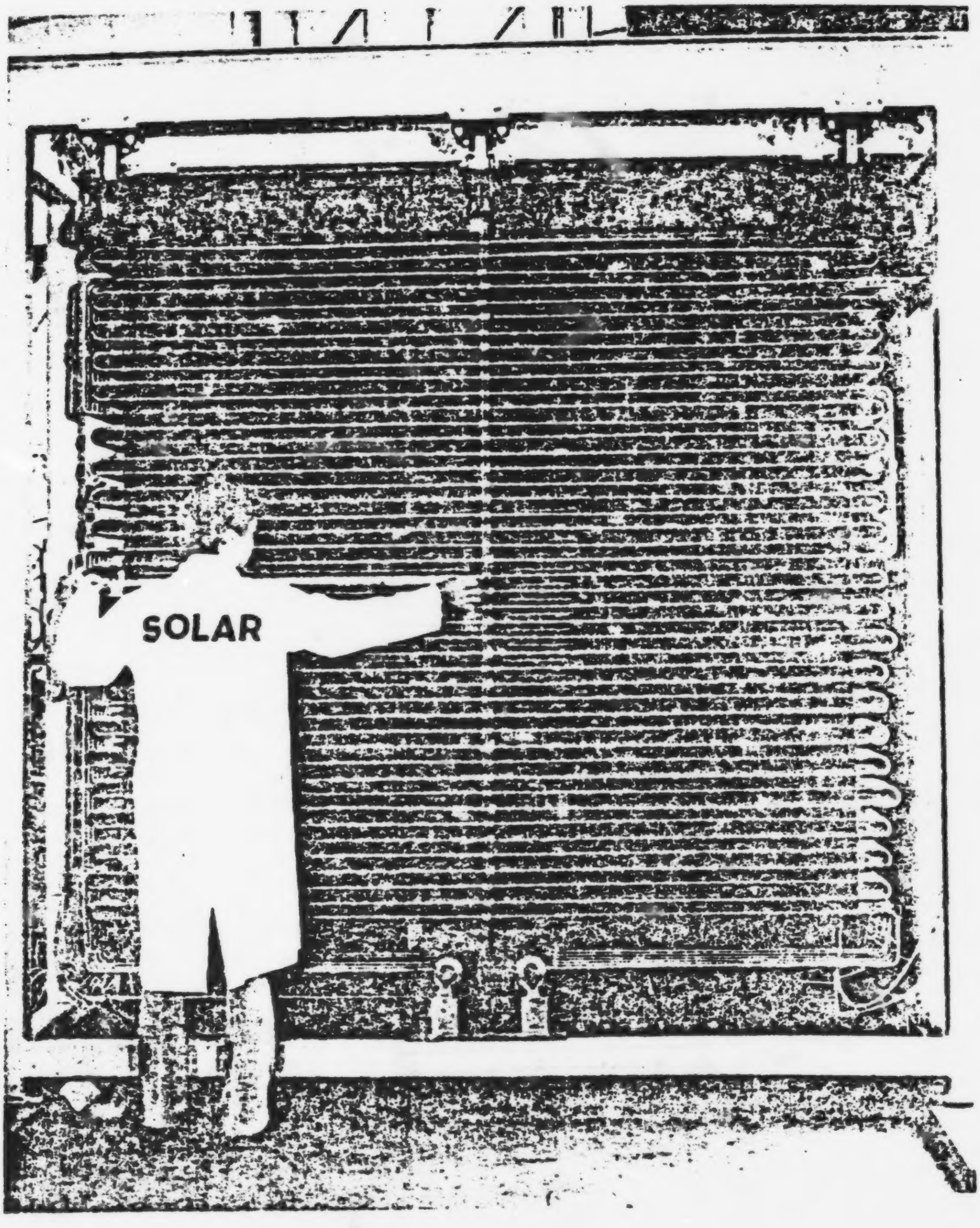


Fig. 4 Combined cycle test boiler module

Steam turbine design considerations

Since steam turbine efficiency directly influences overall steam cycle efficiency, considerable attention has been paid to the design concept evaluation process. A review of commercially available small steam turbines was conducted but ended with the realization that "off-the-shelf" units do not fully satisfy the required performance criteria. The design effort that was subsequently initiated dealt with a variety of mechanical and thermodynamic considerations.

It was soon apparent that there was no practical alternative to the axial flow low-pressure turbine principally because of its ability to handle the rapid increase in volumetric flow characteristics of a condensing turbine system.

The design choices for high pressure turbines were more varied, however. Contra-rotating radial outflow, high speed full admission reaction, high speed radial inflow, and other combinations of reaction and impulse turbines were considered. Two specific turbine designs emerged which exhibited the required degree of performance and mechanical design conservation required for long life and troublefree operation.

One of these, a full admission two-stage high speed axial flow HP geared to an axial LP is shown in Fig. 5. The comparative performance of the units is shown in Fig. 6. Both designs are being considered for specific applications, both marine and industrial.

Control system

The steam cycle is designed to operate as a slave to the gas turbine. This means that boiler pressure will be allowed to vary as gas turbine power varies. This approach eliminates the need for boiler pressure control and permits the use of a full admission steam turbine, thereby also eliminating the losses normally experienced with typical segmented nozzle partial arc admission turbines.

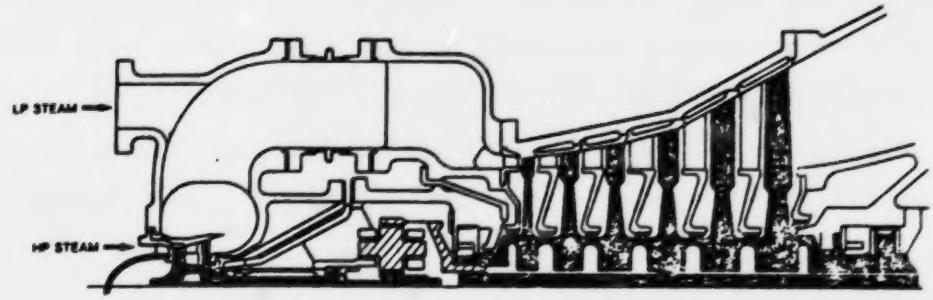

LP STEAM
HP STEAM

Fig. 5 Two-speed steam turbine

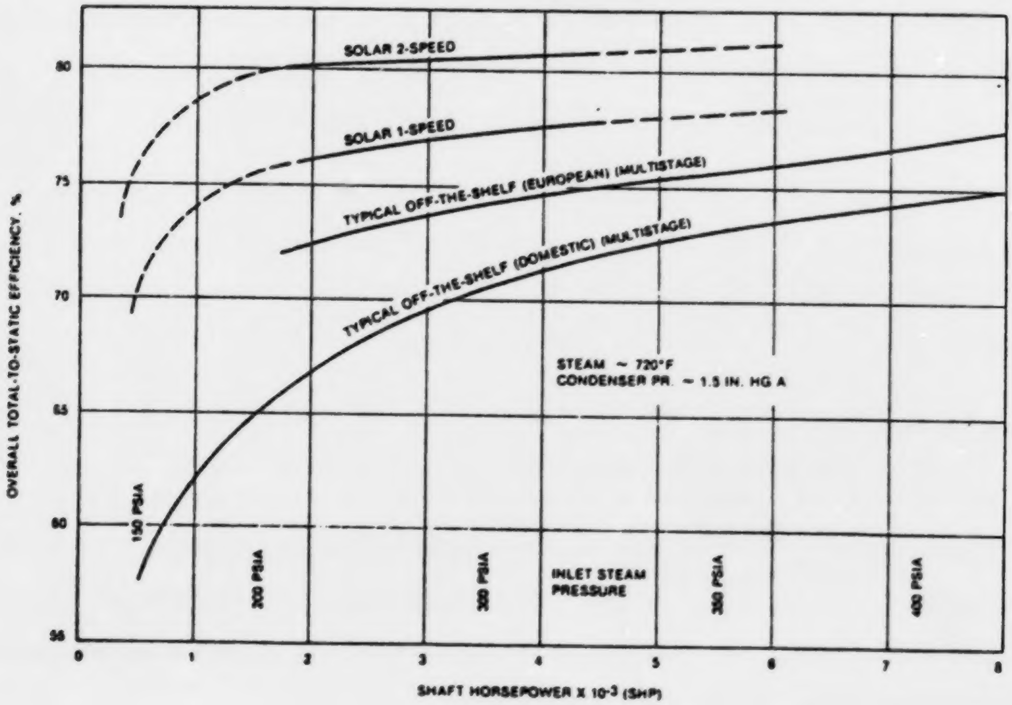


Fig. 6 Condensing steam turbine efficiency

As mentioned earlier, the once-through boiler has comparatively simple control requirements which lead to a simple overall control system. A solid state electronic control module using microprocessor technology has been developed. Two means of boiler control are possible; predictive control of superheater temperature and closed loop control based on boiler gas-side approach temperature.

In the first case, the thermodynamic analysis of the boiler operation indicates that effective control of the feedwater inlet flow rate may be based on enthalpy balance. This means that, with a fixed outlet restriction and a specific input gas enthalpy, the output steam pressure and temperature is directly related to feedwater flow rate. The microprocessor controller uses the predictive energy balance equation to establish an output parameter for use in controlling feedwater control valve position.

In the closed loop operating mode, control is based on the difference between the high pressure steam temperature leaving the boiler and the temperature of the gas turbine exhaust entering the gas side of the boiler. Feedwater flow is regulated to maintain the desired temperature difference.

Fig. 7 is a schematic representation of the microprocessor controller operation.

Feedwater leakage/conservation

As discussed earlier, the design concept chosen for the heat recovery boiler uses corrosion resistant materials for prevention of waterside corrosion and provides for deionization of make-up feedwater to prevent waterside deposits. No chemical feedwater treatment is provided nor are boiler blowdowns required.

The success of this concept depends on a system design which allows an absolute minimum of water/steam leakage from the cycle. The design goal for make-up feedwater rate is twenty gallons per day or less. Obviously, this requirement means that all opportunities for water/steam leakage must be carefully scrutinized and each thoroughly addressed.

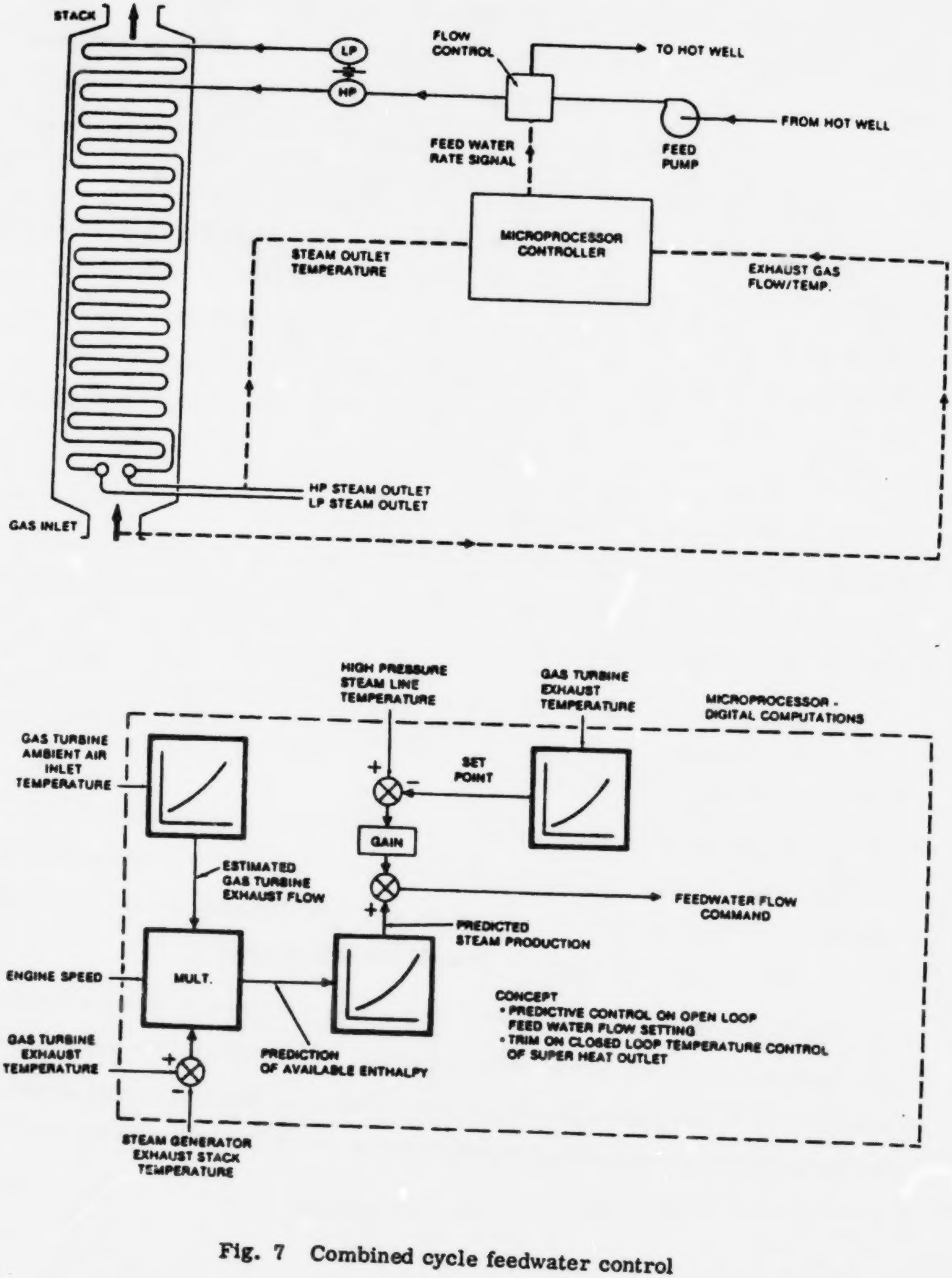

STACK
LP
HP
FLOW CONTROL
TO HOT WELL
FROM HOT WELL
FEED PUMP
FEED WATER RATE SIGNAL
MICROPROCESSOR CONTROLLER
STEAM OUTLET TEMPERATURE
EXHAUST GAS FLOW/TEMP.
HP STEAM OUTLET
LP STEAM OUTLET
GAS INLET
HIGH PRESSURE STEAM LINE TEMPERATURE
GAS TURBINE EXHAUST TEMPERATURE
MICROPROCESSOR - DIGITAL COMPUTATIONS
GAS TURBINE AMBIENT AIR INLET TEMPERATURE
+
−
SET POINT
GAIN
ESTIMATED GAS TURBINE EXHAUST FLOW
FEEDWATER FLOW COMMAND
+
PREDICTED STEAM PRODUCTION
ENGINE SPEED
MULT.
CONCEPT
• PREDICTIVE CONTROL ON OPEN LOOP FEED WATER FLOW SETTING
• TRIM ON CLOSED LOOP TEMPERATURE CONTROL OF SUPER HEAT OUTLET
GAS TURBINE EXHAUST TEMPERATURE
+
−
PREDICTION OF AVAILABLE ENTHALPY
STEAM GENERATOR EXHAUST STACK TEMPERATURE

Fig. 7 Combined cycle feedwater control

The worst offenders in this regard are rotating seals, particularly turbine glands and feed/condensate pump shaft seals. We have been conducting rig testing of extremely low leakage face-type seals for use with the steam turbine. Water pump shaft seal leakage will be prevented by the use of so-called "canned" pumps.

In addition to the goal of low make-up feedwater flow, the low leakage concept provides the added benefits of reducing the capacity requirement of the main condenser vacuum pump and reducing lube oil contamination of system steam and watersides.

System performance

Nominal Mars combined cycle performance is shown in Table 1. The specific fuel consumption rate for the system described, using fuel having a lower heating value of 18,500 Btu/lb is 0.31 lb/HP-hr. This fuel consumption rate is competitive with the most advanced propulsion equipment available in the marine market.

Table 1 Mars combined cycle nominal performance

Simple Cycle (hp), ISO	10,900
Fuel Rate (Btu/hp-hr)	7,770
Combined (hp), ISO	14,735
Combined Fuel Rate (Btu/hp-hr)	5,748
Combined Thermal Efficiency (%)	44.3
Auxiliary Horsepower Required	124
Net Horsepower, ISO	14,611
Net Fuel Rate (Btu/hp-hr)	5,796
Net Thermal Efficiency (%)	43.9
Net Fuel Savings (Btu/hp-hr)	1,974
Net Fuel Savings (%)	25.4
Specific Fuel Consumption (lb/hp-hr)	0.31

Ship installation considerations

Fig. 8 illustrates the Mars combined cycle and indicates package dimensions. As mentioned earlier, the modular approach is appropriate for the system.

A study was conducted to compare machinery size and weight for competitive steam, diesel and combined cycle powerplants. Table 2 compares engine room volume necessary for appropriate arrangements of the three types of machinery. The combined cycle system requires an engine room volume approximately equal to the medium speed diesel and 12 percent smaller than the steam system used for comparison. This comparison was made without using any unusual arrangement techniques. Engine room size and volume could be further reduced by mounting the boiler beside the gas turbine and by using the "folded" arrangement with the powerplant over the propeller shaft.

The weight comparison is shown in Table 3. The compilation includes main engines, reduction gears, distilling plants, propeller and shafting, auxiliary systems (lube oil, fuel, etc.), auxiliary boilers, liquid filling and transfer, ships service generators, engine and propulsion system controls and switchgear. While of paramount importance to the naval ship designer, savings in machinery weight offers benefits to the commercial ship designer also, considering, for example, that engine weight can be exchanged for either cargo weight or fuel weight.

NO.	MAJOR COMPONENTS DESCRIPTION	APPROX. DRY WTS
1	MARS GAS TURBINE & STRUCTURE	23.8 TONS
2	COMBINED REDUCTION DRIVE	8.2 TONS
3	STEAM TURBINE	1.8 TONS
4	STEAM GENERATOR	25.8 TONS
5	DRIVE SHAFT WITH COUPLINGS	0.6 TONS
6	CONDENSER	9.0 TONS
7	SEPARATE LUBE SYSTEM MODULE	1.5 TONS

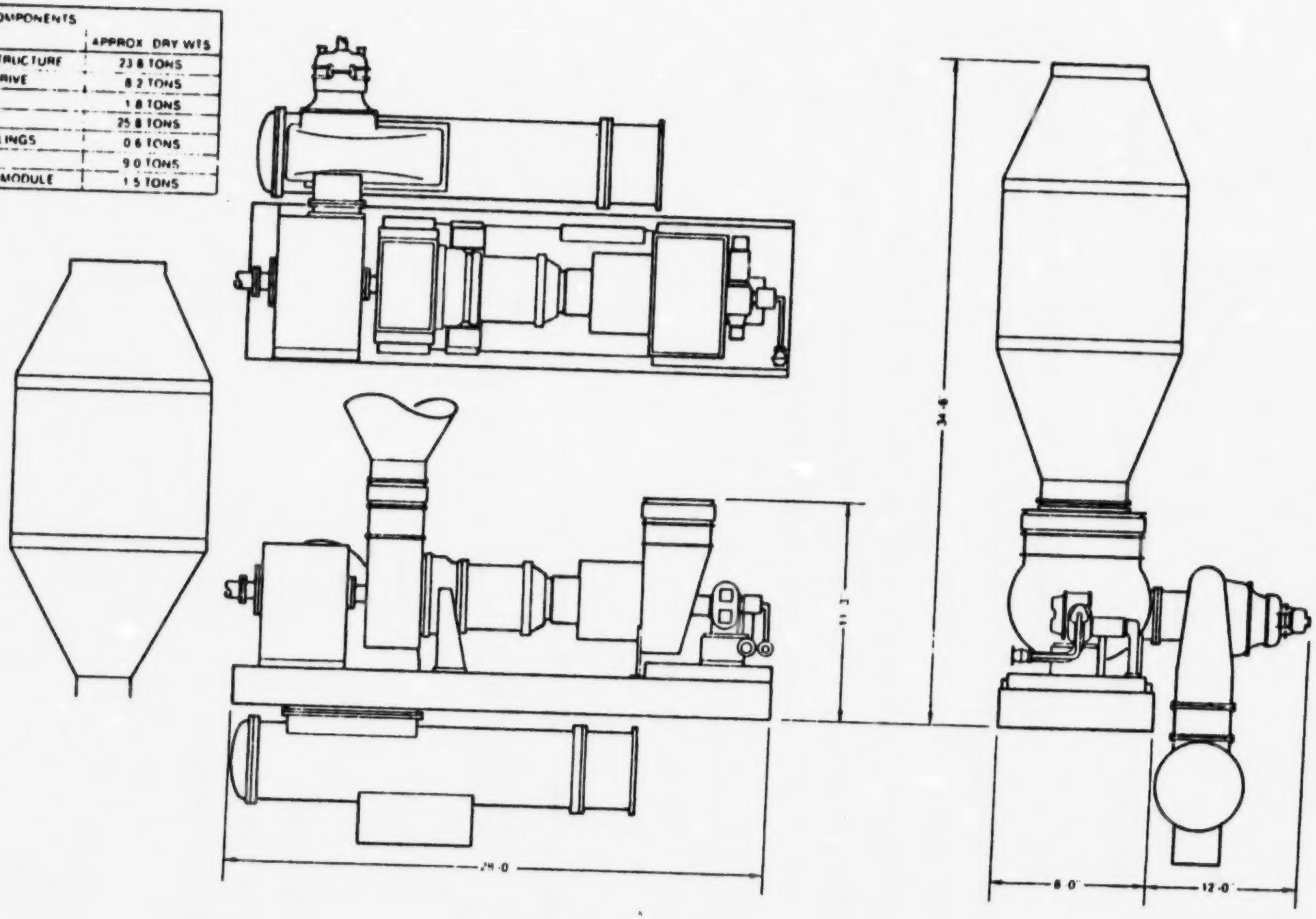


Fig. 8 Mars combined cycle outline drawing

Table 2 Machinery arrangement space comparison for 14,000 HP plants

	Steam (Mariner Class)	Medium Diesel	Mars Combined Cycle
Engine Room Length (ft)	50	49	48
Engine Room Volume (ft^3)	196 M	176 M	174 M
Boiler Height (ft)	22	-	23
Engine Length (ft)	12	31	28
Engine Width (ft)	22	12.5	16
Engine Height (ft)	14	17	17

Table 3 Summary of weights for machinery

	Total (tons)	Tons Heavier	% Heavier
Mars Combined Cycle	396.4	-	-
Medium Speed Diesel	442.0	45.6	11.5
Steam 850/950 Two Heater	716.5	320.1	80.8

Conclusion

The combined cycle propulsion system described herein offers solutions to problems commonly encountered with typical steam and diesel propulsion systems through the practical application of advanced technology. A compact and highly efficient propulsion system has been developed which offers fuel economy, installation simplicity and ease of operation. Shipboard maintenance requirements may be reduced to a very low level by utilizing modular maintenance and exchange techniques.

The order for the first shore-based installation of a combined cycle based on this design was received in November 1978. Experience gained in that installation will be applied to the marine system.

As the economic realities of the continuing rise in fuel costs become more and more apparent, the need for more efficient propulsion systems will also become more apparent. The combined cycle powerplant can satisfy that need.

Table 2 Machinery arrangement space comparison for 14,000 HP plants

	Steam (Mariner Class)	Medium Diesel	Mars Combined Cycle
Engine Room Length (ft)	50	49	48
Engine Room Volume (ft^3)	196 M	176 M	174 M
Boiler Height (ft)	22	-	23
Engine Length (ft)	12	31	28
Engine Width (ft)	22	12.5	16
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References

1 Mills, R.G., "Greater Ship Capability and Energy Saving with Combined Cycle Machinery," Naval Engineers Journal, Volume 89, No. 5, October 1977.

2 Wardall, R.M. and Doorly, E.E., "Current Prospects for Efficient Combined Cycles for Small Gas Turbines." Paper presented at the ASME Gas Turbine Conference, Houston, Spring 1976.

3 Marron, H.D. and Carleton, R.S., "The Gas Turbine Waste Heat Recovery System and the U.S. Navy," ASME Paper 78-GT-170.

4 "The Impact of a High Performance 14,000 HP Marine COGAS Plant", Seaworthy Engine Systems Report 037-R1.5, October 22, 1976.

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Design Aspects of an Advanced 14,000 HP Marine Combined Cycle Plant

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2200 Pacific Highway, P.O. Box 80966, San Diego, California 92138

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DESIGN ASPECTS OF AN ADVANCED 14,000 HP MARINE COMBINED CYCLE PLANT

by

Norman L. MacIntyre

Presented to

American Society of Mechanical Engineers
Twenty-Fourth Annual Gas Turbine Conference
San Diego, California

March 11-15, 1979

SOLAR TURBINES INTERNATIONAL
An Operating Group of International Harvester

ABSTRACT

A discussion of the design aspects for a marine combined cycle propulsion system utilizing a dual pressure cycle is presented. The system discussed is based on the 10,000 hp class Mars gas turbine recently introduced to the industry. Certain hardware aspects and recent testing are discussed.

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INTRODUCTION

One of the most immediate effects of the recent dramatic increase in fuel costs has been a search for ways to improve the efficiency of power systems. The obvious fact is that more efficient systems consume less fuel. The increase in fuel costs has affected the economic payback situation with the result that added costs for more efficient equipment can be justified within normally acceptable financial guidelines. Thus, forward thinking owners and operators have become more conscious of total operating costs, rather than acquisition costs only, when making powerplant selections for new shipbuilding programs.

Accordingly, an advanced combined cycle powerplant is being developed whose system design is intended to meet certain specific requirements. The principal of these are:

- High efficiency
- Simplicity
- Complete automation
- Minimal package size
- Modular assembly

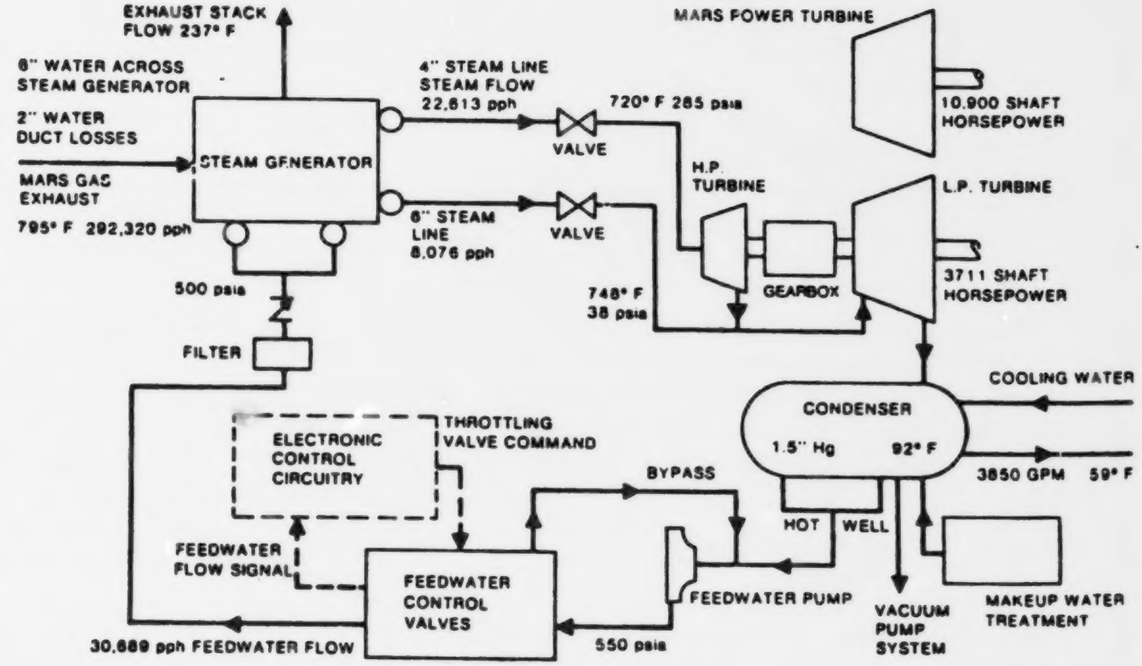
BACKGROUND

CYCLE SECTION

The requirement for a system having the highest practicable efficiency dictated a review of the basic thermodynamic restraints on the combined gas turbine (Brayton) and steam (Rankine) cycles. Accordingly, a comprehensive evaluation of cycle combinations and steam conditions was conducted with a view toward maximizing overall combined cycle efficiency. This evaluation and the rationale involved has been reported by Wardall and Doorly (1) and Mills (2). A schematic of the system is shown in Figure 1.

The evaluation resulted in the selection of a dual pressure steam cycle as superior to the commonly used single pressure cycle. Figures 1 and 2 illustrate the fact that with the dual pressure cycle, the Rankine cycle working fluid (water/steam) temperature more closely follows the gas turbine exhaust gas temperature through the boiler than with a single pressure cycle.

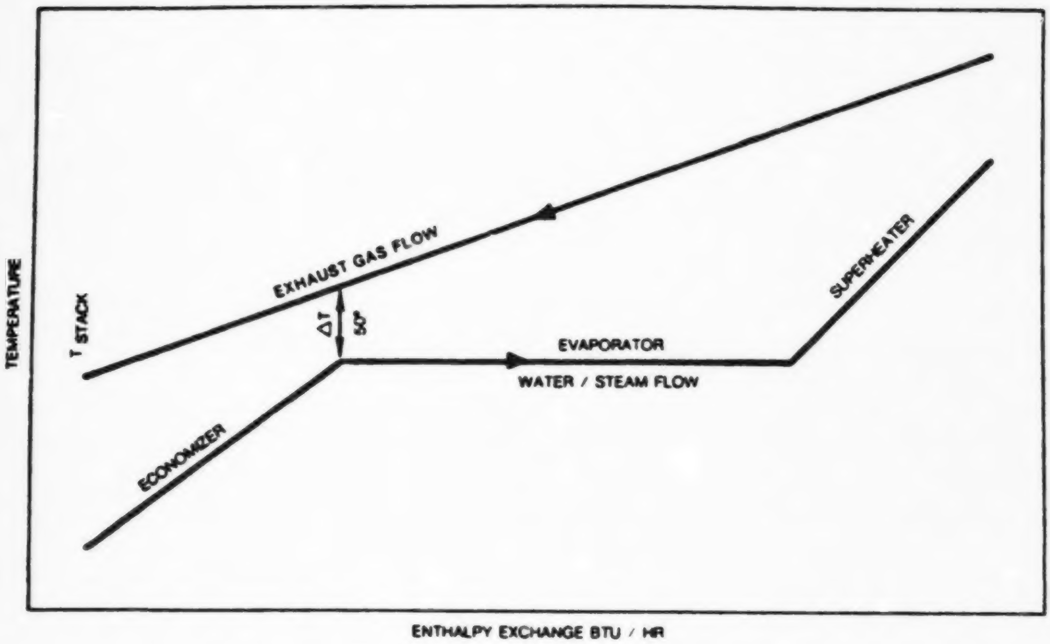
Higher [illegible]
[illegible]


EXHAUST STACK FLOW 237° F
6" WATER ACROSS STEAM GENERATOR
2" WATER DUCT LOSSES
STEAM GENERATOR
MARS GAS EXHAUST
795° F 292,320 pph
500 psia
FILTER
4" STEAM LINE STEAM FLOW 22,613 pph
6" STEAM LINE 8,076 pph
VALVE
VALVE
720° F 285 psia
748° F 38 psia
MARS POWER TURBINE
10,900 SHAFT HORSEPOWER
H.P. TURBINE
GEARBOX
L.P. TURBINE
3711 SHAFT HORSEPOWER
COOLING WATER
CONDENSER
1.5" Hg
92° F
3850 GPM
59° F
HOT WELL
ELECTRONIC CONTROL CIRCUITRY
THROTTLING VALVE COMMAND
BYPASS
FEEDWATER FLOW SIGNAL
FEEDWATER CONTROL VALVES
FEEDWATER PUMP
VACUUM PUMP SYSTEM
MAKEUP WATER TREATMENT
30,689 pph FEEDWATER FLOW
550 psia

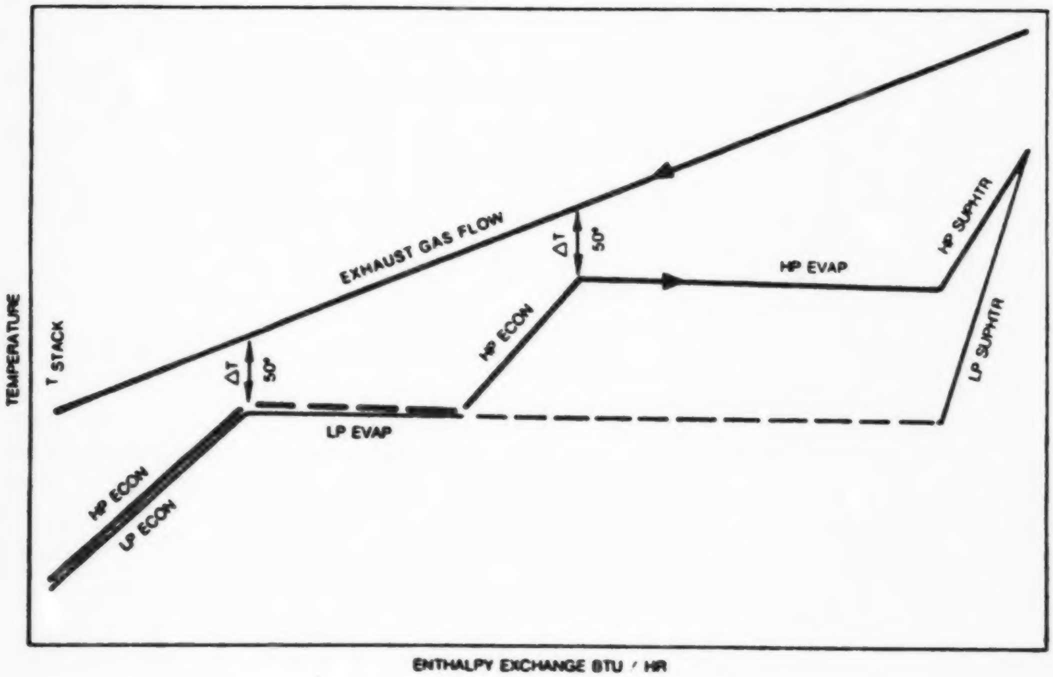
NOTES:

1. **Auxiliary power is included (3%).**
2. **The data shown are for the power plant sizing case with the gas turbine at ISO conditions with steam generator losses included.**

Figure 1. Mars Combined Cycle


TEMPERATURE
T STACK
EXHAUST GAS FLOW
ΔT
50°
EVAPORATOR
WATER / STEAM FLOW
SUPERHEATER
ECONOMIZER
ENTHALPY EXCHANGE BTU / HR

a. Single pressure cycle


TEMPERATURE
T STACK
EXHAUST GAS FLOW
ΔT
50°
ΔT
50°
HP EVAP
HP SUPHTR
HP ECON
LP SUPHTR
LP EVAP
HP ECON
LP ECON
ENTHALPY EXCHANGE BTU / HR

b. Dual pressure cycle

Figure 2. Pinch Point Diagrams

In review, it is generally recognized that heat transfer area is dictated by pinch point temperature differences (ΔT in Figure 2, a and b). For tube areas not to become infinitely large, the pinch point temperature difference must have some reasonable positive value (technical-economic studies usually indicate 30-50°F for most installations). Comparative calculations readily indicate that, if the gas turbine exhaust temperature is in the 800°F region, it is not possible to reduce the stack temperature to the desired level of 235°F with a single pressure system. This is the basic reason for choosing a dual-pressure cycle. In practical terms, the dual pressure cycle results in a 12 percent gain in power output over a single pressure cycle with the same 50°F pinch point temperature difference.

BOILER DESIGN

Initially, the desire to use a dual pressure steam cycle seemed contrary to the goal of overall simplicity. It was clear that an innovative approach to boiler design was required. The solution to this problem lay in the "once-through" boiler.

The once-through boiler has no drums and therefore requires no level controls. It is composed of economizer, evaporator and superheater sections, connected in series. The change of state from water to steam takes place in the evaporator tubes directly, such that only water enters the evaporator section and only steam continues to the superheater. The once-through boiler is shown schematically in Figure 3, arranged for dual pressure steam generation.

STEAM TURBINE DESIGN

Since the beginning of the combined cycle evaluation study, considerable attention has been directed toward the steam turbine. The efficiency of the turbine directly influences the efficiency of the overall cycle. With this in mind an industry search was conducted to determine the availability of high efficiency small turbines. It soon became apparent that generally available turbines did not meet the required performance criteria. In addition to efficiency goals, these criteria included simplicity for ease in startup and shutdown and ability to withstand variations in steam temperature resulting from changes in load and firing rate.

The design effort that was subsequently initiated to develop a high efficiency reliable small steam turbine investigated a variety of mechanical and thermodynamic considerations.

It was soon apparent that there was no practical alternative to the axial flow low-pressure (LP) turbine, principally because of its ability to handle the rapid increase in volumetric flow associated with a condensing turbine system.

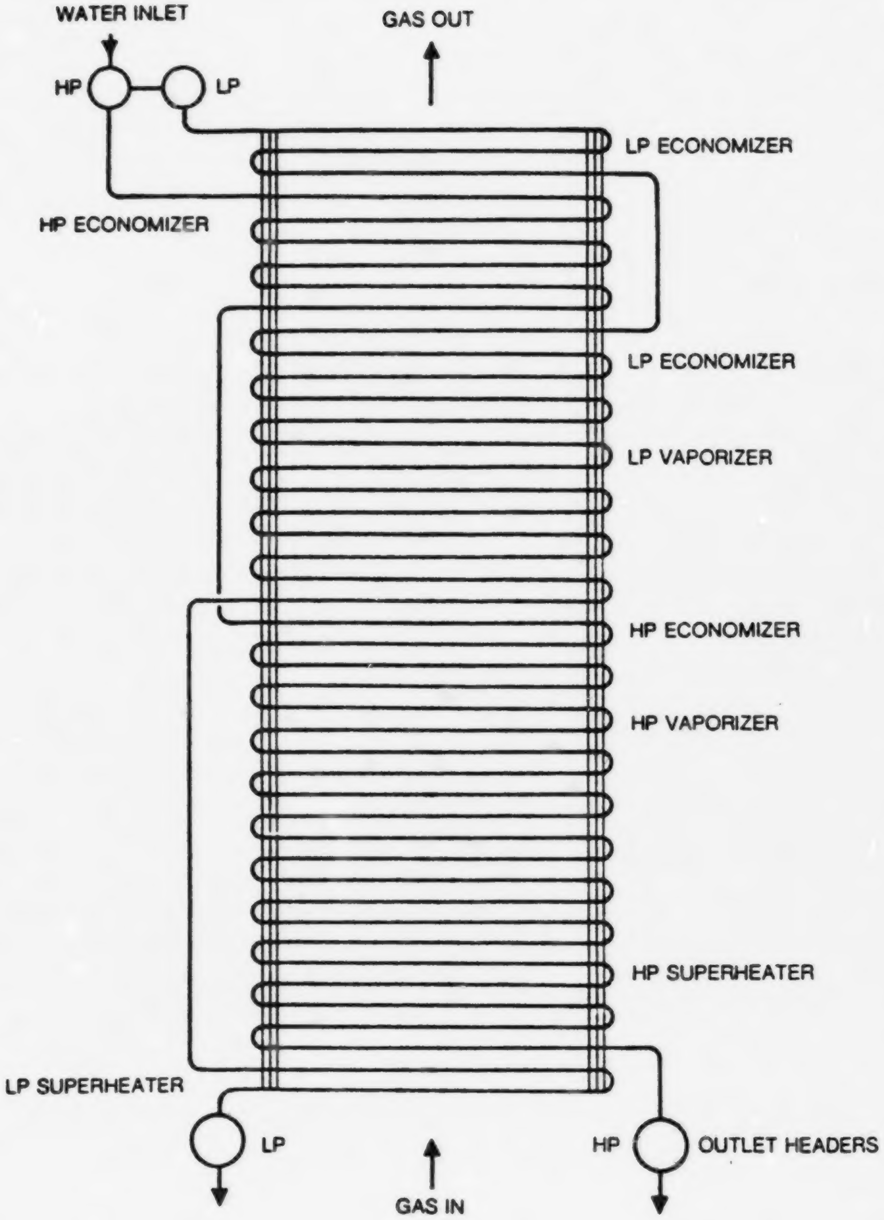

WATER INLET
GAS OUT
HP
LP
LP ECONOMIZER
HP ECONOMIZER
LP ECONOMIZER
LP VAPORIZER
HP ECONOMIZER
HP VAPORIZER
HP SUPERHEATER
LP SUPERHEATER
LP
HP
OUTLET HEADERS
GAS IN

Figure 3. Mars Combined Cycle Boiler

The design choices for high pressure (HP) turbines were more varied, however. Contra-rotating radial outflow, high speed full admission reaction, high speed radial inflow and other combinations of reaction/impulse turbines were considered. One design which emerged is shown in Figure 4. This unit has a high speed (58,000 rpm) full admission two-stage axial HP section geared to an axial flow LP (10,700 rpm) section.

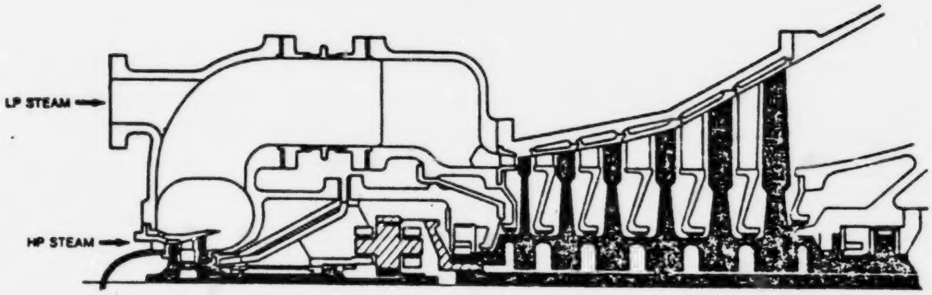


Figure 4. Two-Speed Steam Turbine

CONTROL PHILOSOPHY

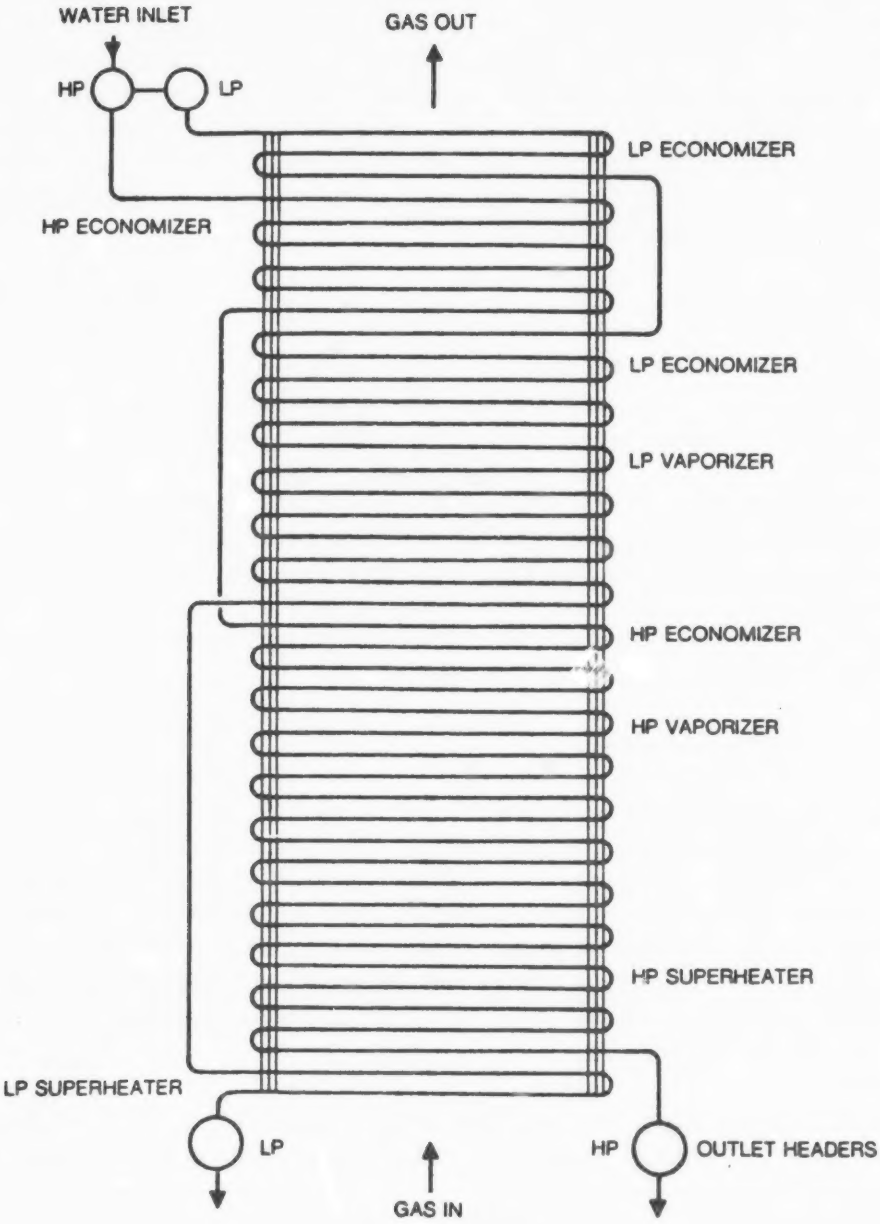
The steam cycle is designed to operate as a slave to the gas turbine. Boiler pressure will be allowed to vary with gas turbine power. The full admission turbine will operate without a throttle, dependent only on boiler pressure. This control scheme eliminates the requirement for boiler pressure control and also eliminates throttling losses normally experienced when turbine power is controlled independently.

The minimal control requirements of the once-through boiler permit a simple overall control system. Figure 5 is a schematic representation of the microprocessor control strategy. As is shown, feedwater flow is regulated as a function of boiler stack temperature, gas turbine exhaust temperature, gas turbine speed, ambient air temperature and boiler HP outlet steam temperature.

The concept employs open loop predictive control of the feedwater flow setting and closed loop temperature control trimming of the superheater outlet. The microprocessor controller was developed in 1976 and 1977. Development testing has been extremely successful. The response and stability of the boiler when controlled by the microprocessor is discussed in part in the following section.

RECENT TESTING

An aggressive testing program has been conducted to develop the concepts just described into hardware components which are reliable and perform as desired. Certain elements of this testing program are presented in the following sections.


WATER INLET
GAS OUT
HP
LP
LP ECONOMIZER
HP ECONOMIZER
LP ECONOMIZER
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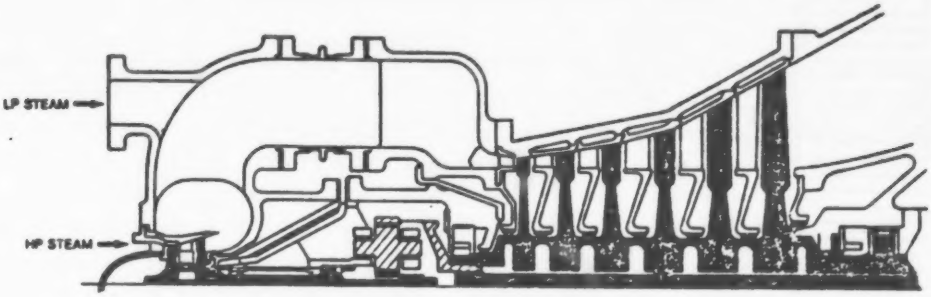

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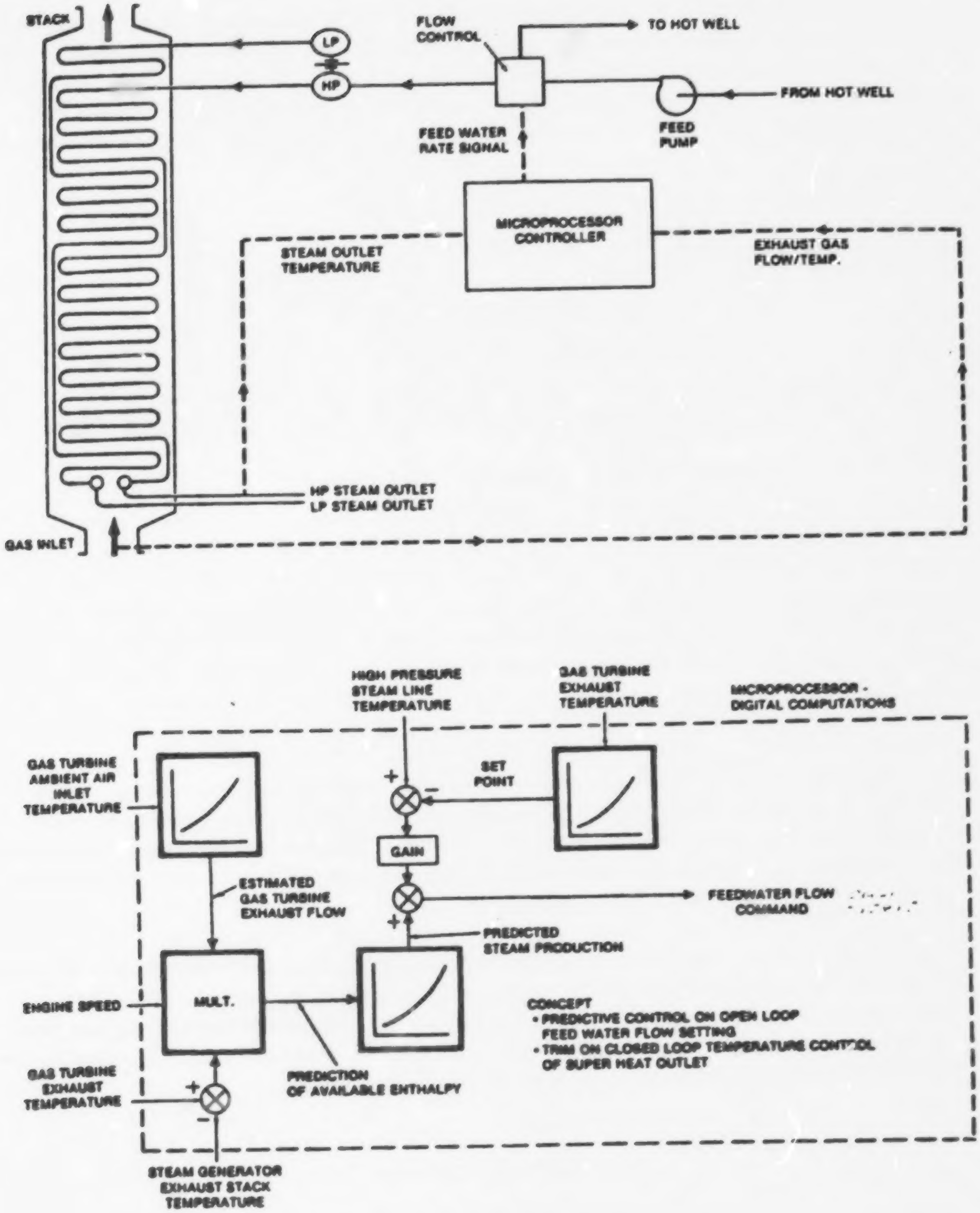

STACK
LP
HP
FLOW CONTROL
TO HOT WELL
FROM HOT WELL
FEED PUMP
FEED WATER RATE SIGNAL
MICROPROCESSOR CONTROLLER
STEAM OUTLET TEMPERATURE
EXHAUST GAS FLOW/TEMP.
HP STEAM OUTLET
LP STEAM OUTLET
GAS INLET
HIGH PRESSURE STEAM LINE TEMPERATURE
GAS TURBINE EXHAUST TEMPERATURE
MICROPROCESSOR - DIGITAL COMPUTATIONS
GAS TURBINE AMBIENT AIR INLET TEMPERATURE
+
−
SET POINT
GAIN
ESTIMATED GAS TURBINE EXHAUST FLOW
FEEDWATER FLOW COMMAND
+
PREDICTED STEAM PRODUCTION
ENGINE SPEED
MULT.
CONCEPT
• PREDICTIVE CONTROL ON OPEN LOOP FEED WATER FLOW SETTING
• TRIM ON CLOSED LOOP TEMPERATURE CONTROL OF SUPER HEAT OUTLET
GAS TURBINE EXHAUST TEMPERATURE
+
−
PREDICTION OF AVAILABLE ENTHALPY
STEAM GENERATOR EXHAUST STACK TEMPERATURE

Figure 5. Combined Cycle Feedwater Control

STEAM GENERATOR MODULE AND DYNAMIC MODEL

The boiler module shown in Figure 6, a one-sixth "slice" of a full-sized boiler, has been used for experimental purposes since it was built in early 1977. This module contains full-sized steam generation circuits and only differs from a complete boiler in the number of circuits used. The module has been used for a variety of testing purposes, involving both the steam/water side and the gas side.

Concurrent with the construction of the test boiler module, a nonlinear dynamic model of the boiler was developed. The model is being used to study boiler dynamic response to inputs and disturbances. It is also used to develop and validate control concepts for the boiler in different applications.

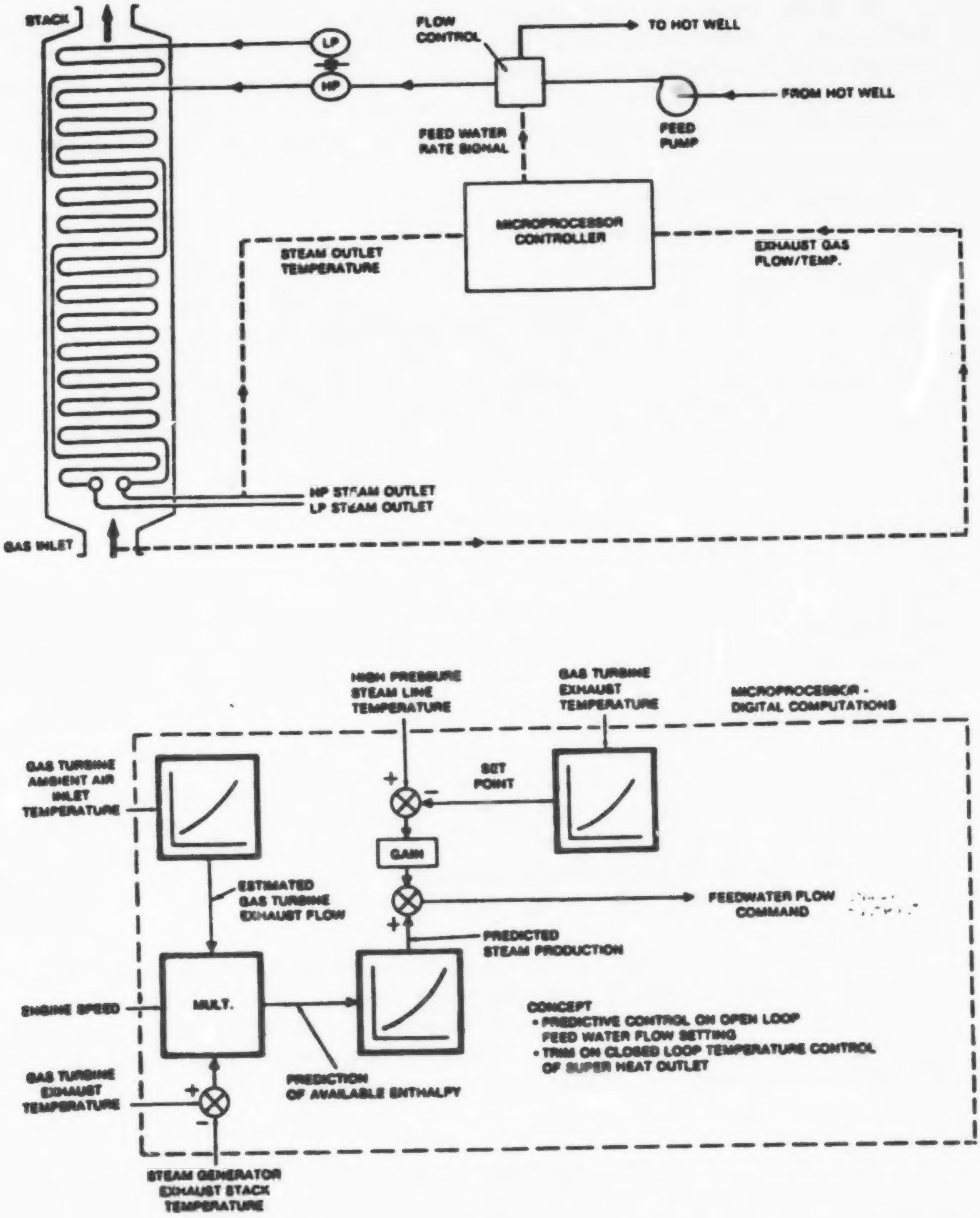
The model consists of twenty-nine first order, nonlinear, ordinary differential equations with associated algebraic constraints. It is a "three-phase-moving boundary" type model in which the boiler is divided into variable volumes consistent with the phases of steam in various tube segments (i.e., economizer, evaporator, superheater). Model equations are derived from fundamental conservation relations through the control volume approach.

Preliminary comparisons of model and test module response to steps in feedwater flow show very good low frequency agreement. Figures 7 and 8 show inlet flow steps, temperature response, and pressure response at the HP and LP exits for the module and model. The correspondence of low frequency behavior is evident. Ongoing model validation efforts will involve the use of systematic techniques to identify model parameters (heat transfer coefficients, etc.) from test module data.

The model is presently being used to study control concepts proposed for combined cycle systems that use the once-through boiler. A numerically linearized version of the model is also being used to determine controller compensation and gain settings.

GAS SIDE CLEANING

The use of liquid fuel in the conventional gas turbine combustion system results in the production of carbonaceous soot as a by-product of local inefficiencies in the combustion process. Through the development and application of improved fuel injectors such as the airblast atomizer, soot levels have been drastically reduced even to a point where such emissions are low enough to avoid any visible exhaust smoke. It has also been demonstrated experimentally that the future generation of advanced, pre-mixed combustion systems are capable of operation with essentially zero levels of soot production. Unfortunately, these systems are several years away from application in production marine gas turbines. Therefore, the problem of steam generator gas side contamination by combustor-generated soot must be addressed during the design phase of the integrated gas turbine combined cycle system.


STACK
LP
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MICROPROCESSOR - DIGITAL COMPUTATIONS
GAS TURBINE AMBIENT AIR INLET TEMPERATURE
+
−
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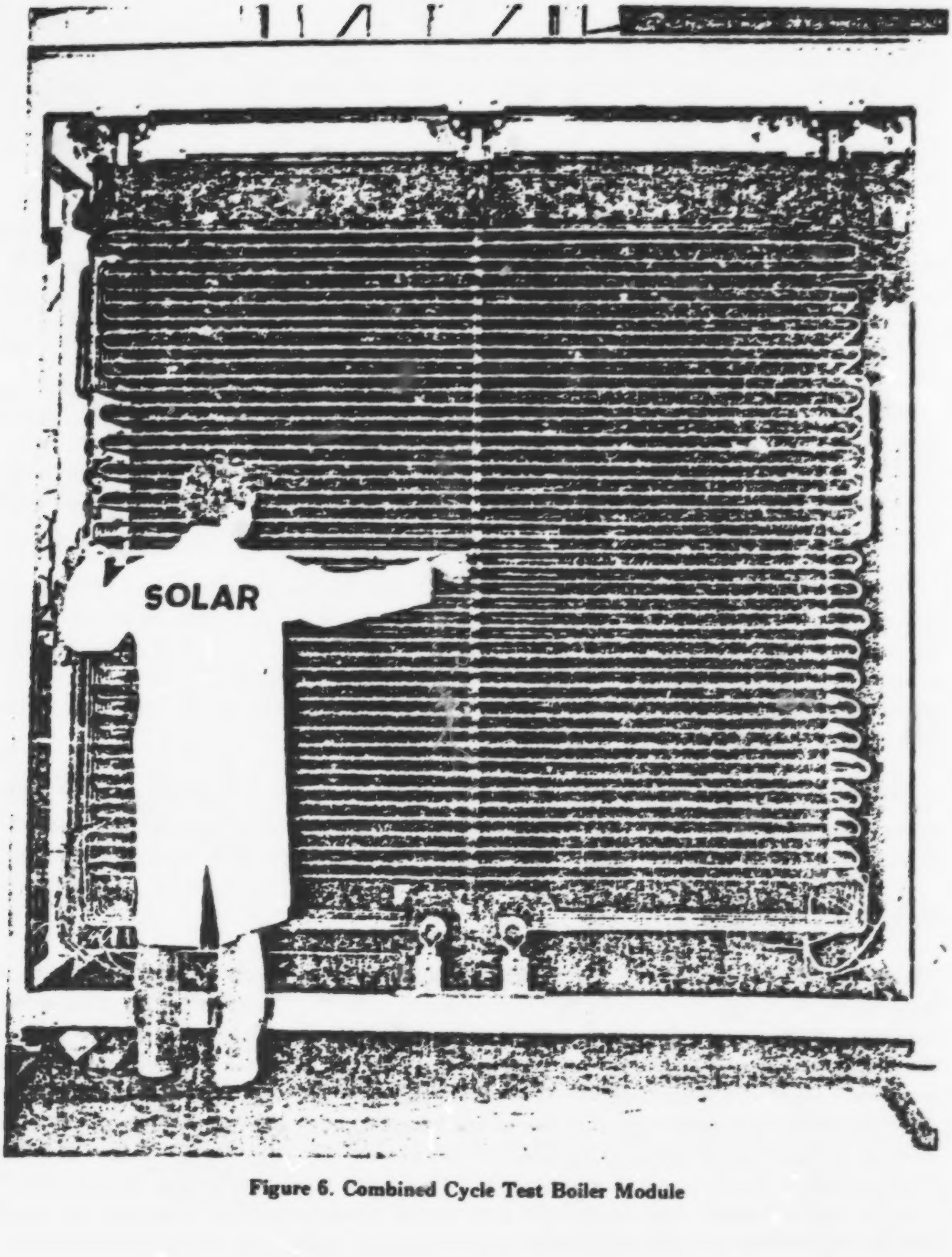
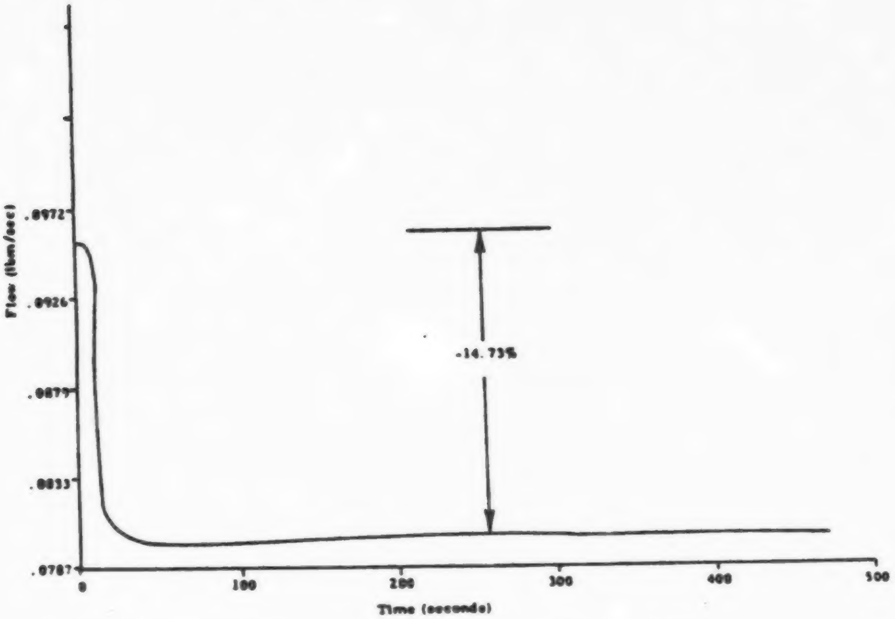
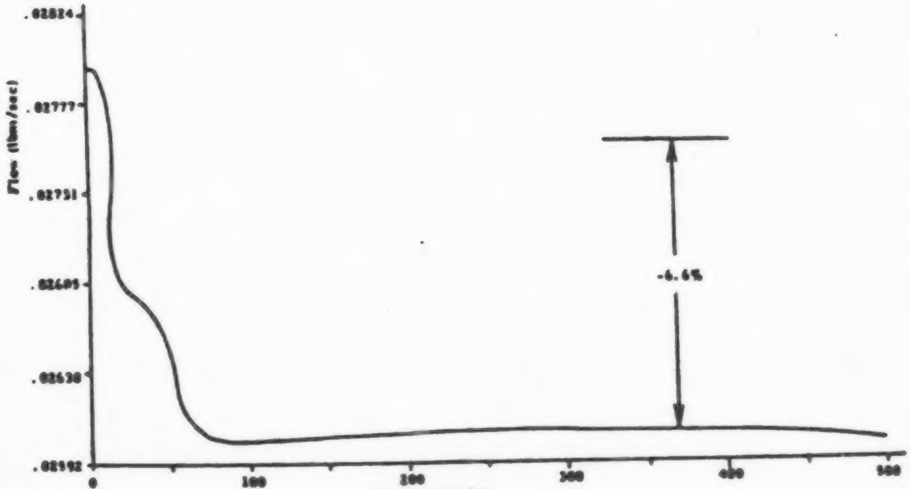


Figure 6. Combined Cycle Test Boiler Module

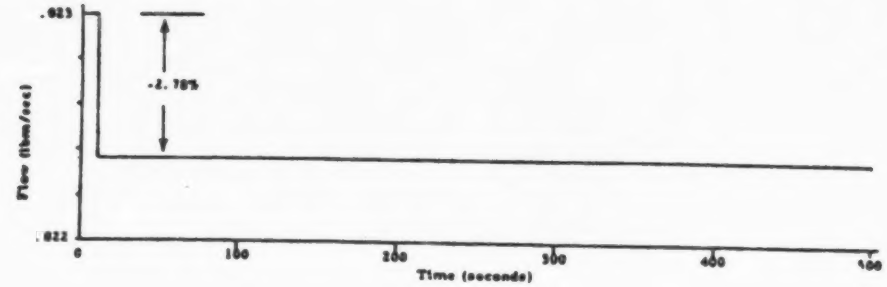


a. Disturbance to High Pressure Feed Flow



b. Disturbance to Low Pressure Feed Flow

Figure 7. Boiler Module Response



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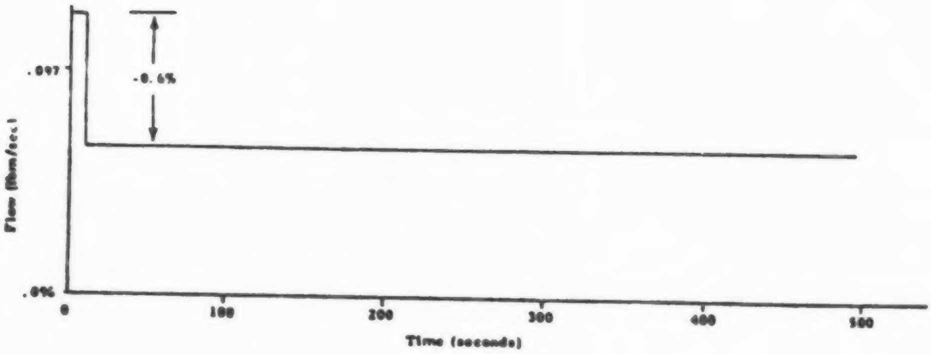
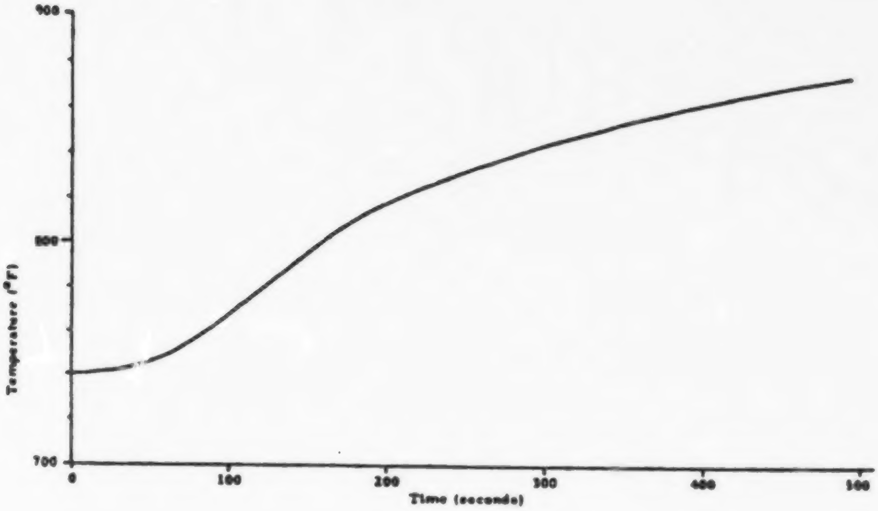
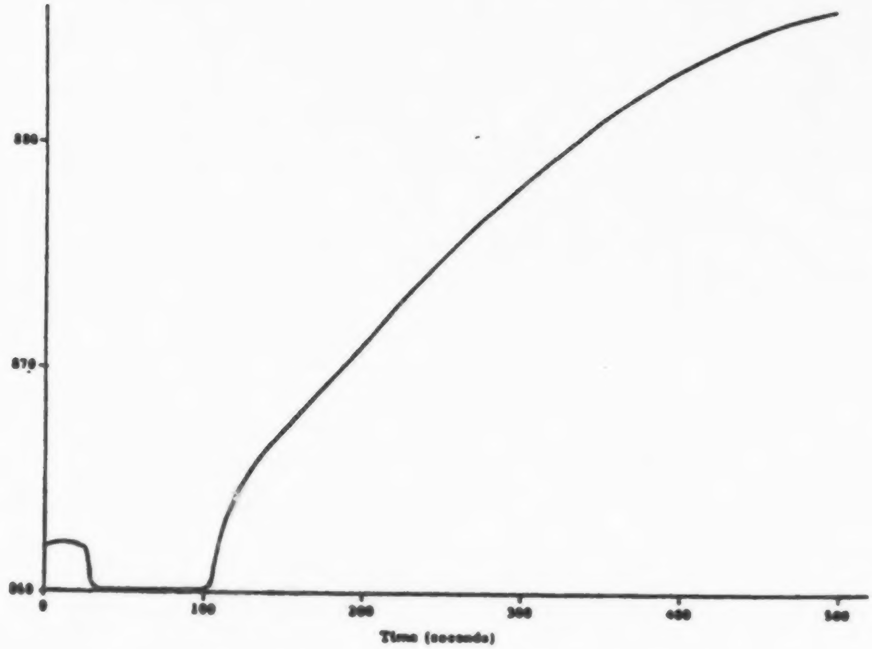


b. Disturbance to Low Pressure Feed Flow

Figure 8. Dynamic Model Response

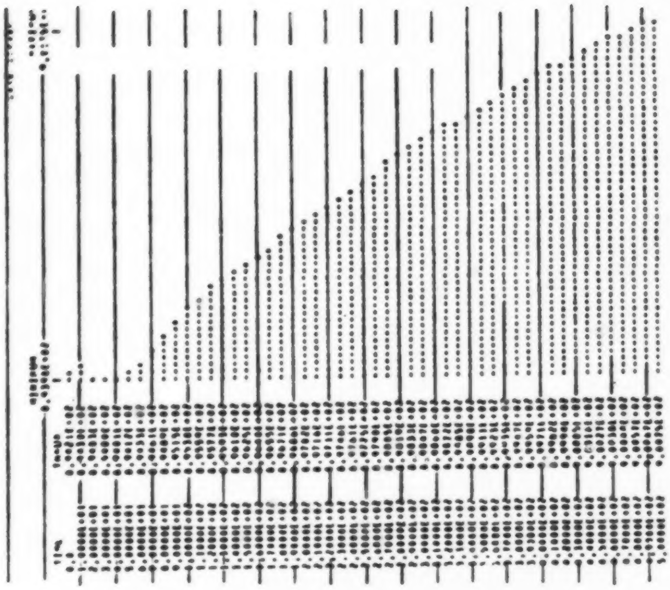


c. Response of High Pressure Outlet Temperature

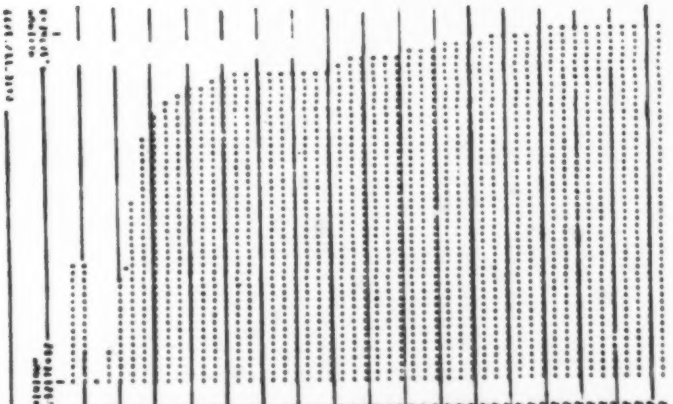


d. Response of Low Pressure Outlet Temperature

Figure 7. Boiler Module Response, Contd

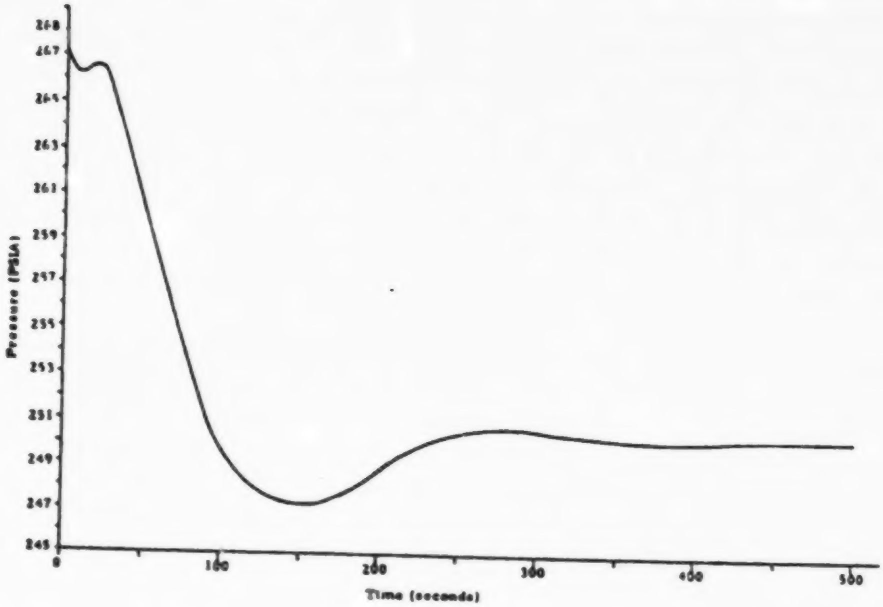


c. Response of High Pressure Outlet Temperature

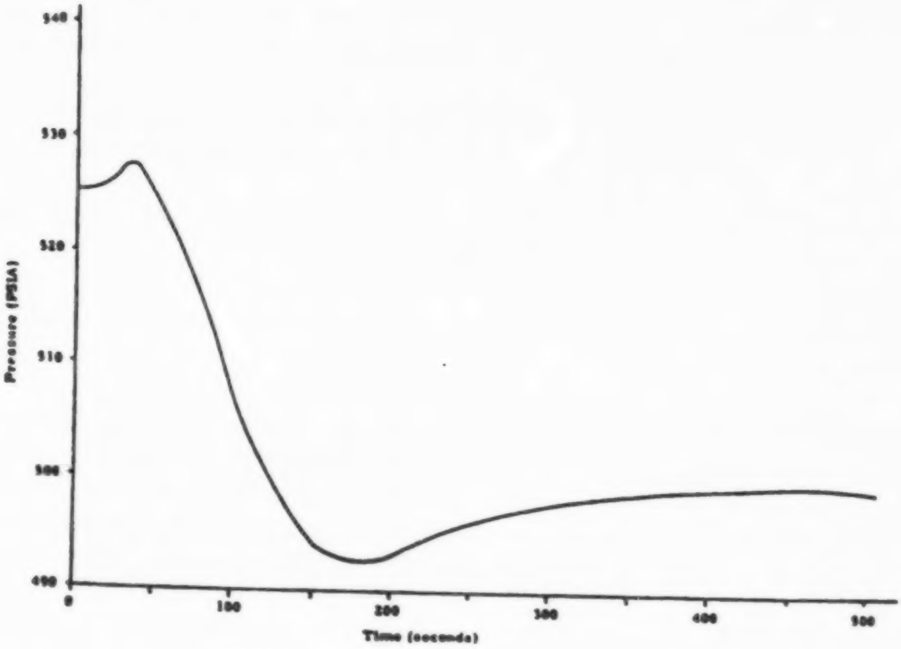


d. Response of Low Pressure Outlet Temperature

Figure 8. Dynamic Model Response, Contd

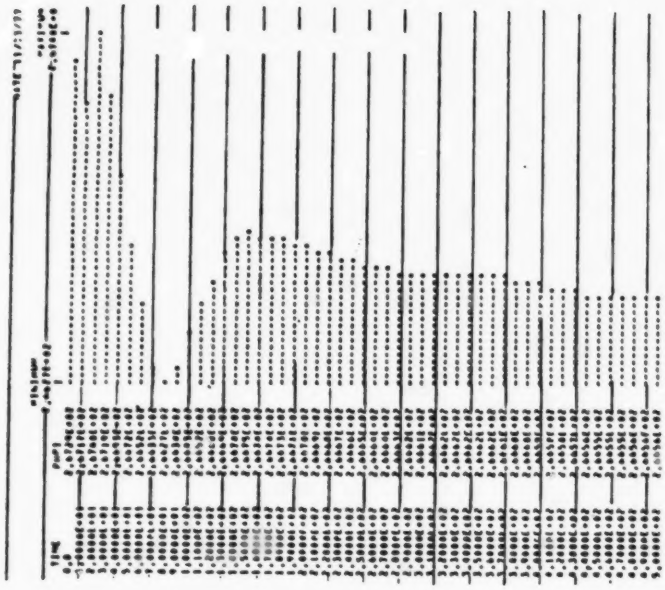


e. Response of High Pressure Outlet Pressure

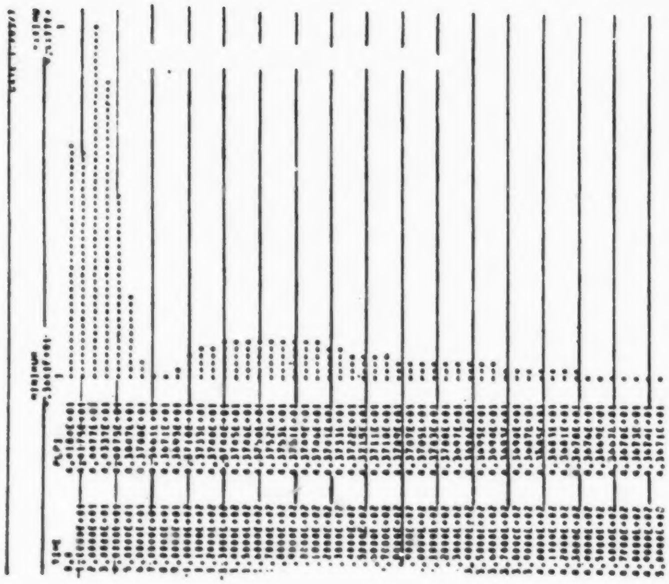


f. Response of Low Pressure Outlet Pressure

Figure 7. Boiler Module Response, Contd



e. Response of High Pressure Outlet Pressure



f. Response of Low Pressure Outlet Pressure

Figure 8. Dynamic Model Response, Contd

Gas side fouling has an impact on the overall combined cycle gas turbine system in the areas of performance, safety and maintenance. Build-up of deposits on the heat transfer surfaces acting as an insulating layer reduces heat transfer rates and hence steam generator effectiveness. Extreme levels of deposits can produce fin bridging with drastic reductions in the effective heat transfer areas. The increase in gas side pressure drop resulting from fouling also has a direct deterimental effect on the gas turbine output because of the increase in backpressure. Excessive gas side soot deposits may pose a safety hazard if unburned fuel is dumped into the turbine exhaust, as, for example, during a start cycle light-off failure. In addition, fouling of the gas side surfaces can impact operating personnel levels, depending upon the cleaning method selected.

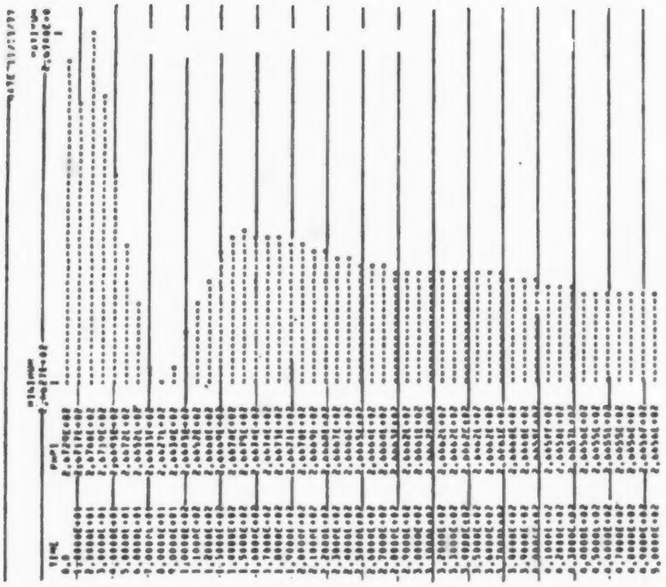
Conventional auxiliary cleaning methods such as soot blowers have well-recognized disadvantages, particularly with the advanced compact matrix type of steam generators previously described. Self-cleaning by dry firing the steam generator has been selected as the primary technique for gas side cleaning purposes. This decision is based on the requirement for an effective method that does not require additional space-utilizing cleaning equipment or additional operational personnel and on the considerable background and understanding in the area of steam generator gas side fouling and cleaning developed from internally funded and U.S. Navy sponsored research programs (3).

The soot fouling and self-cleaning characteristics of the combined cycle steam generator were evaluated during a recent test program conducted with a Solar Centaur turbine generator set. The arrangement is shown in Figure 9. The Centaur turbine generator set is on the right and the steam generator module on the left.

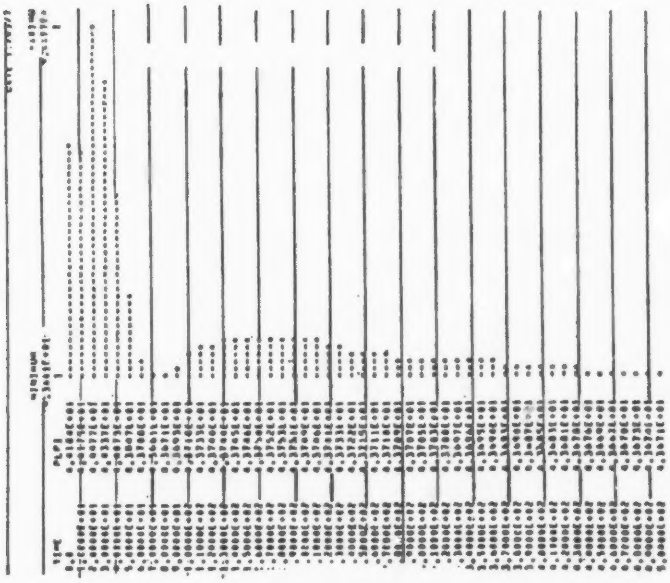
A total of 105 hours were run on No. 2 Diesel fuel at loads varying between 2800-2900 kW. Start cycles were included.

The exhaust opacity varied between 8-10 percent during the test run which is equivalent to a Von Brand smoke number in the 35-40 range and a SAE smoke number of approximately 52-58. A test with an impingement sampler gave an absolute soot loading of 0.02 g/m^3. Exhaust unburned hydrocarbons were monitored at between 1.0 and 4.0 ppm during the test.

The total deterioration in the boiler module effectiveness over the length of the test was approximately 7.0 percent with a test starting point at a deterioration of 1.4 percent from the "as built" conditions. The initial deterioration that had taken place from the "as built" condition is thought to be mainly due to fin oxidation resulting from earlier testing programs. The test was conducted at constant boiler gas side mass flow and the pressure drop increased from a starting value of 4.6 inches of water to approximately 5.8 inches of water at the completion of the test. At the completion of the test period the deterioration rate was close to zero and the indications from the performance calibration points were that equilibrium had perhaps been achieved.



e. Response of High Pressure Outlet Pressure



f. Response of Low Pressure Outlet Pressure

Figure 8. Dynamic Model Response, Contd

Gas side fouling has an impact on the overall combined cycle gas turbine system in the areas of performance, safety and maintenance. Build-up of deposits on the heat transfer surfaces acting as an insulating layer reduces heat transfer rates and hence steam generator effectiveness. Extreme levels of deposits can produce fin bridging with drastic reductions in the effective heat transfer areas. The increase in gas side pressure drop resulting from fouling also has a direct deterimental effect on the gas turbine output because of the increase in backpressure. Excessive gas side soot deposits may pose a safety hazard if unburned fuel is dumped into the turbine exhaust, as, for example, during a start cycle light-off failure. In addition, fouling of the gas side surfaces can impact operating personnel levels, depending upon the cleaning method selected.

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Figure 9. Module Fouling Test Arrangement

Subsequent to the steam generator module tests on the Centaur engine a self-cleaning test was run on the dry boiler. The total time between the completion of the fouling test and the next engine start was 255 hours. A pre-dryout calibration showed that the boiler performance had recovered from the 8 percent deterioration at the completion of the fouling test to a 3 percent deterioration level. The cause of this improvement is now known, however a considerable amount of flaking of the soot deposits was noted as the boiler was idle between runs. This calibration point was followed by 1.5 hours of dry boiler operation at the full load engine conditions. After an overnight cool-down period the boiler was recalibrated and the results showed that dry running had reduced the total deterioration to within 0.5 percent of the level at the beginning of the fouling test.

The total rig and engine background developed in the areas of soot fouling and self-cleaning investigations strongly indicate that self-cleaning by dry operation is a viable choice for the primary cleaning technique. The many variables involved in the complete characterization of the self-cleaning phenomenon are not, however, completely recognized at this time and a backup system has been devised for use if required.

FULL SCALE TEST FACILITY

Figure 10 shows the new combined cycle test facility being constructed for full scale system development testing. Figure 11 shows the arrangement of components within the test cell.

The facility includes on-line automatic data acquisition and processing equipment to provide project personnel with immediate data analysis.

SUMMARY

The nominal system performance is shown in Table 1, and a suggested package arrangement is illustrated in Figure 12.

The basic design concepts outlined are, in some aspects, a departure from accepted steam system design practice in the past. This has been the result of a thorough analysis of the thermodynamic and practical systems barriers experienced with these systems, backed up by extensive laboratory and module testing. The knowledge gained to date has borne out the validity of this new design approach.

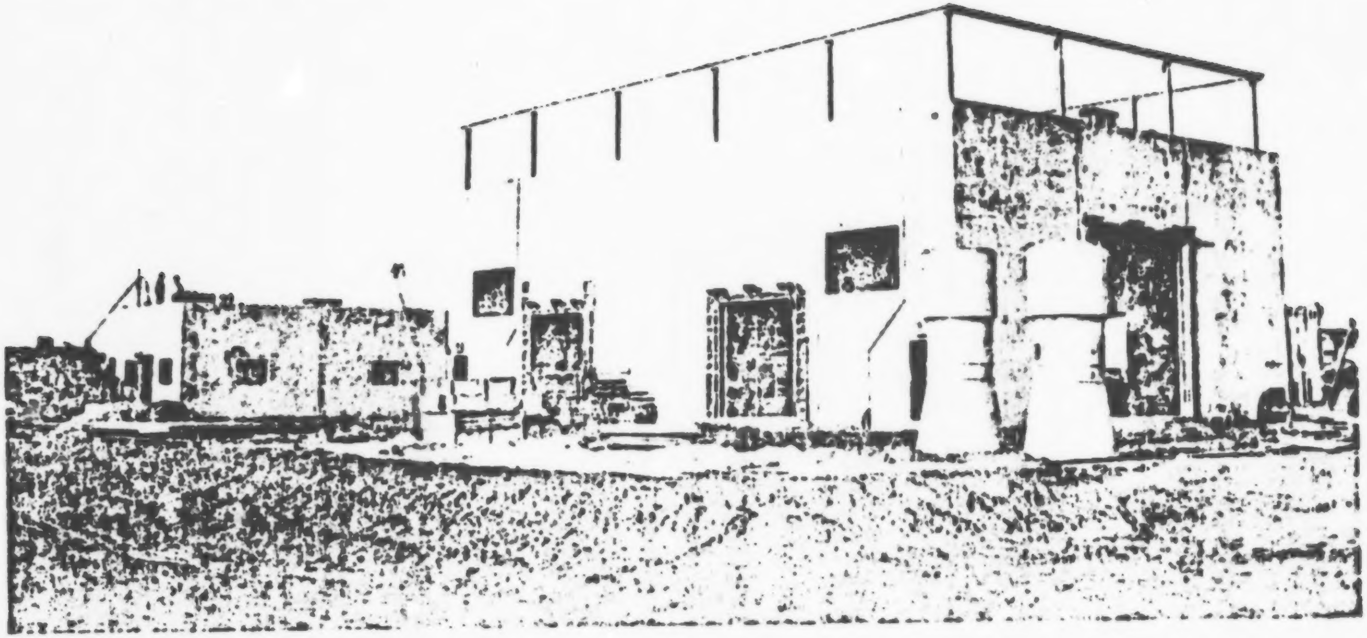


Figure 10. Combined Cycle Test Facility

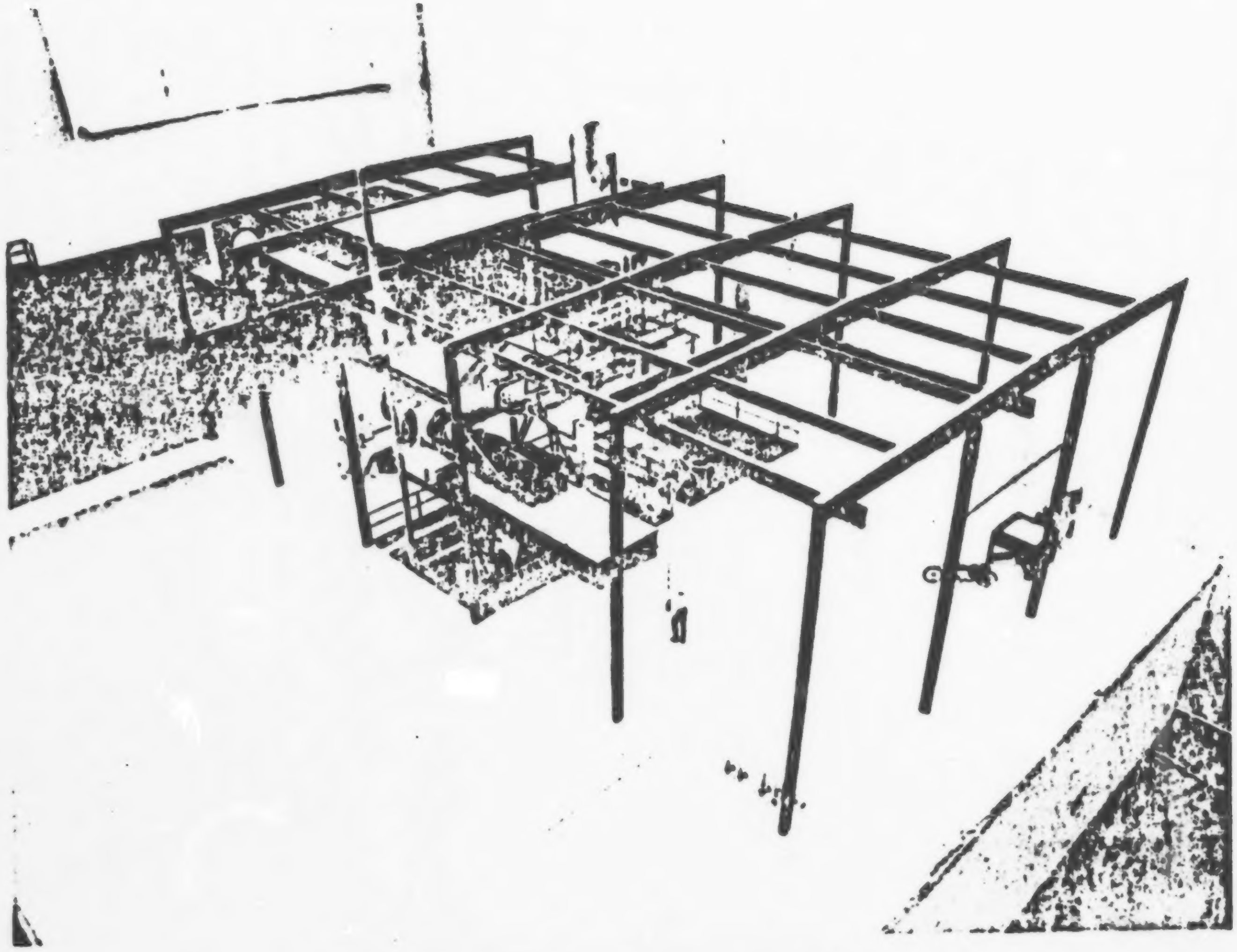
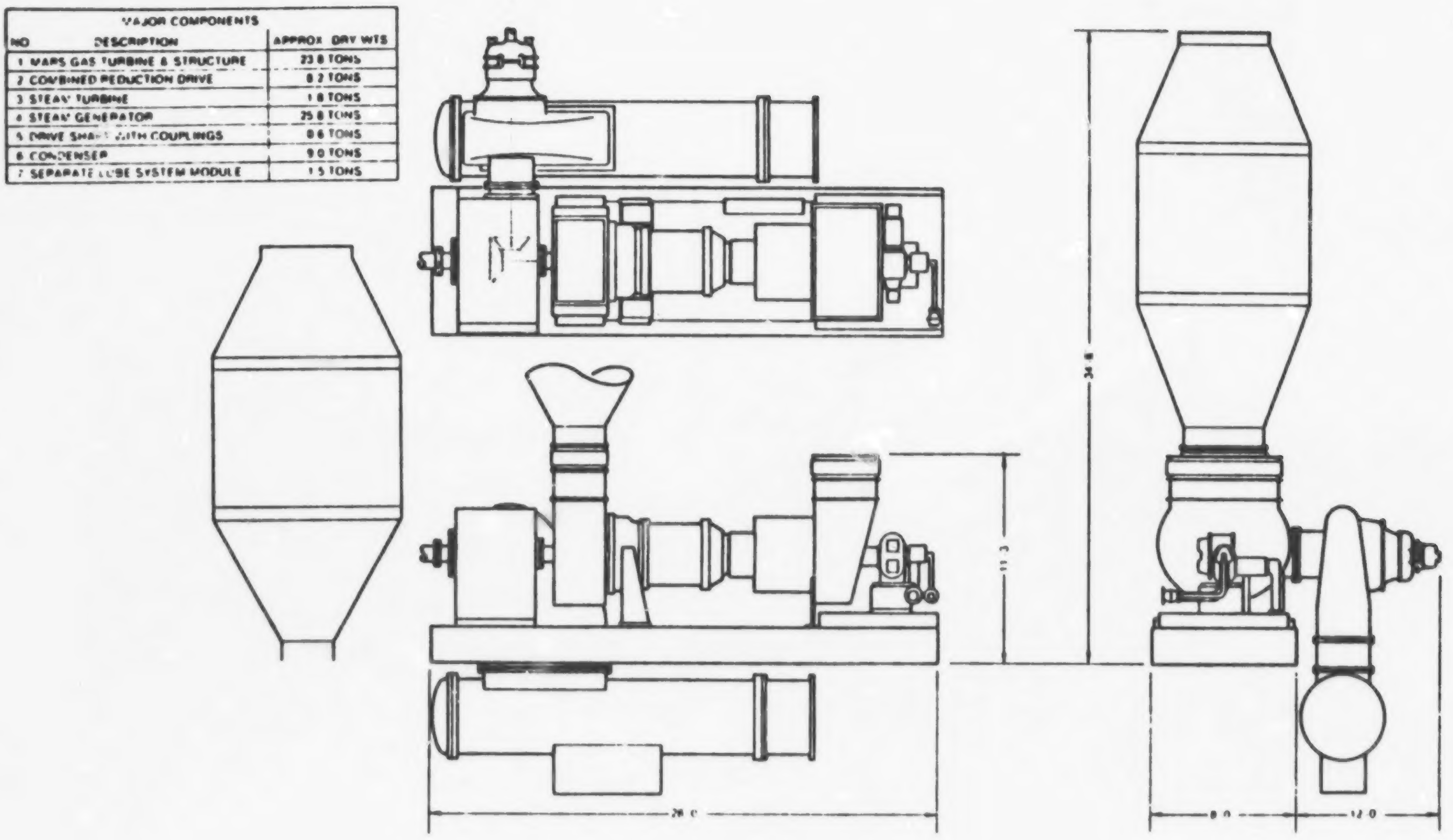


Figure 11. Combined Cycle Test Facility Model



MAJOR COMPONENTS		
NO	DESCRIPTION	APPROX DRY WTS
1	MARS GAS TURBINE & STRUCTURE	23.8 TONS
2	COMBINED REDUCTION DRIVE	8.2 TONS
3	STEAM TURBINE	1.8 TONS
4	STEAM GENERATOR	25.8 TONS
5	DRIVE SHAFT WITH COUPLINGS	0.6 TONS
6	CONDENSER	9.0 TONS
7	SEPARATE LUBE SYSTEM MODULE	1.5 TONS

Figure 12. Mars Combined Cycle Outline Drawing

Table 1. Mars Combined Cycle Nominal Performance

Simple Cycle (hp), ISO	10,900	
Fuel Rate (Btu/hp-hr)	7,770	
Combined (hp), ISO	14,735	
Combined Fuel Rate (Btu/hp-hr)	5,748	
Combined Thermal Efficiency (%)	44.3	
Auxiliary Horsepower Required	124	
Net Horsepower, ISO	14,611	34% HP gain
Net Fuel Rate (Btu/hp-hr)	5,796	
Net Thermal Efficiency (%)	43.9	
Net Fuel Savings (Btu/hp-hr)	1,974	
Net Fuel Savings (%)	25.4	
Specific Fuel Consumption (lb/hp-hr)	0.31	

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77-GT-106

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Design Considerations for Heat Recovery System for DD-963 Class Ship

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Mem. ASME

The advent of high-speed gas-turbine-powered naval vessels requires that new concepts in energy generation and conservation be employed. This paper describes the heat recovery system designed for the U.S. Navy Spruance Class destroyers as applied to the ships' service generator sets. Among the topics discussed in the paper are the water and steam system, controls, heating surface selection and configuration, water treatment, gas-side fouling, gas-side pressure drop, and heat transfer. The thermal power recovered is used to perform heating, distillation plant operations, and galley services, thereby reducing the fuel consumption required to perform these functions. The use of separately fired boilers to perform the required tasks would significantly increase the fuel consumption rate.

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Design Considerations for Heat Recovery System for DD-963 Class Ship

Y. KATZ J. L. BOYEN

CONCEPT OF SYSTEM

High-speed gas turbine powered naval vessels should have auxiliary equipment compatible with the high performance requirements of the ship. Total weight of fuel and equipment is critical and must be held to a minimum level consistent with the ship's purpose and mission. Since the ship's fuel carrying capability governs its range, it is vital that fuel consuming equipment, such as the main propulsion turbines, ship service turbo-generators, and auxiliaries, be as efficient as possible.

The DD-963 ships are equipped with three 2500-kw ship service turbo-generators burning the same grade of fuel oil as are the propulsion turbines. Two of the three sets are capable of carrying the electrical load when at sea and usually one set can carry the load when in port. Therefore, heat recovery from the exhaust of the gas turbo-generators can be used to generate steam required for heating, galley and distillation service.

The gas turbine-generators operate at a full load efficiency of approximately 24 percent. At full load, the turbine exhaust heat recovery capability is about 16,000 lb/hr based on generating steam at 100 psig from feedwater at 180 F. This is a heat recovery rate of approximately 16,500,000 Btu/hr which is recovered without the expenditure of any additional fuel oil. The overall efficiency now goes from 24 to 68 percent. Heat recovery and overall efficiency can be substantially increased by using higher temperature feedwater and by the use of an economizer section on the boiler. Under the above conditions, overall efficiency could be increased to approximately 78 percent.

HEAT RECOVERY SYSTEM

The heart of the energy recovery system is the heat recovery device used to reclaim waste heat from the turbine exhaust. The heat recovery equipment used must be practical, rugged, compact, and serviceable.

There are several options which could be considered for heat or energy recovery. These are gas-air recuperator, water heating, both low and high pressure, steam generation, or organic fluid systems. In order to match the proposed heat utilization system on the DD-963, a medium pressure (100-psig) steam system was selected. This system will adequately operate the ship's heating system, galley and distillation plant. The other forms of heat recovery were not considered feasible for the applications.

There are numerous industrial applications of gas turbine heat recovery systems. However, marine and naval applications are limited, because, in the past, most ships were conventional steam powered. The DD-963 program consists of 30 ships, each ship having three heat recovery systems, making a total of 90 boilers. Several ships are now in service and thousands of operating hours have been accumulated on the heat recovery systems. Therefore, it is now appropriate that the heat recovery system be examined in some detail.

HEAT RECOVERY BOILER DESIGN

In considering alternate systems for the heat recovery system, once-through, spillover and forced recirculation were studied. The final decision was for a forced recirculation system due to its inherent stability, ease of control, and reliability. Natural circulation systems were also considered, but due to space and weight constraints and dry running requirement, this concept was eliminated.

The forced recirculation system is illustrated in Fig. 1. The water reservoir is a vertical vessel 24 in. in diameter with a cyclone separator unit on the top. The approximate water capacity up to the control level is 40 gal. Feedwater is introduced into this vessel as a function of water level change as detected by differential

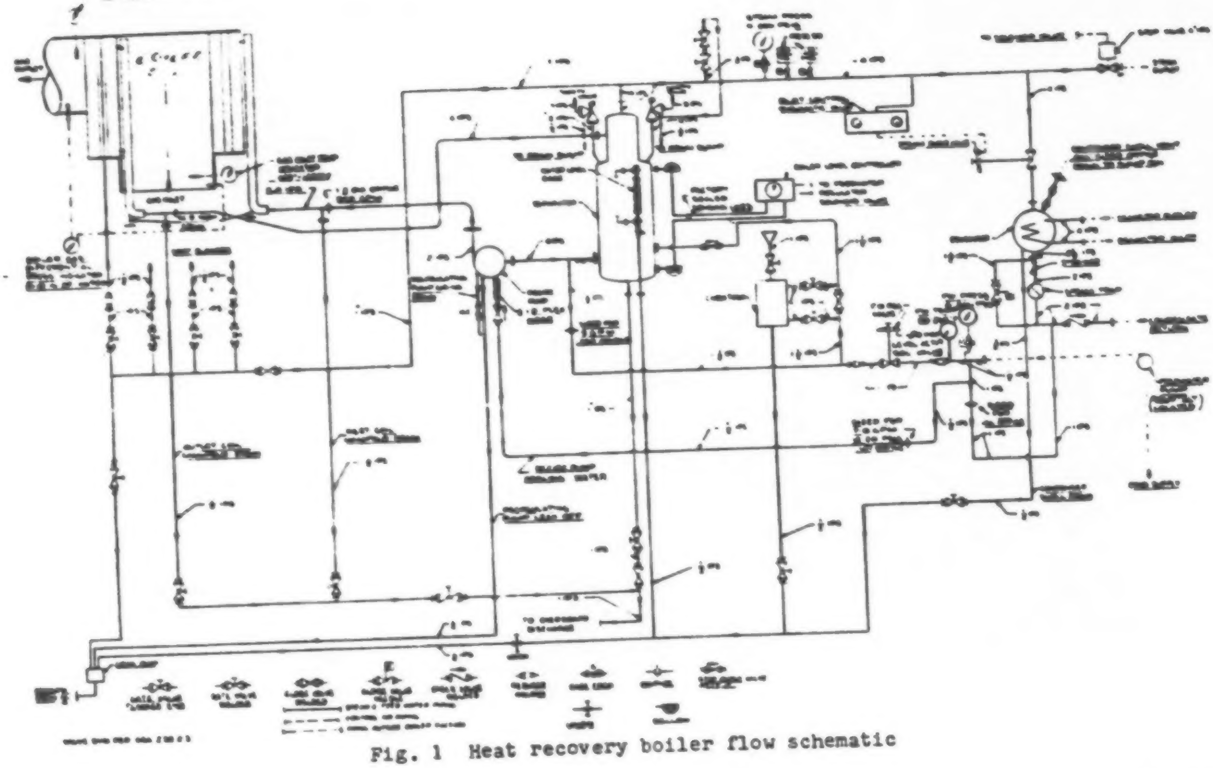

Fig. 1 Heat recovery boiler flow schematic

level controller which senses changes in water level in the vessel as compared to a constant head in a sealed water leg. The recirculation pump takes water out of the reservoir and circulates it through the boiler coils where some water is evaporated. The mixture of water and steam passes into the cyclone separator where the water is separated out and falls back into the vessel. Steam passes out through the top of the separator by way of steam drying centrifugal baffles. Steam quality is approximately 99.8 percent. The recirculation rate is 144 gpm which is ten times the evaporation rate at the design point of 7000 lb/hr and approximately six times the maximum evaporation rate of 12,500 lb/hr.

The arrangement of the water recirculation system, feedwater and steam piping and components are depicted in Figs. 2 and 3, respectively.

The hot gas from the turbine enters the boiler at the bottom and enters the coil inner core. From here it flows radially outward through the coil gas spaces, and out all around the coil-casing annulus and then finally exits from the casing.

Baffles, circular in form and curved toward the coil, guide the hot gas toward the coil, reducing turbulence in the center and, in consequence, reducing the gas pressure drop by about 2.0 in. w.c. At the gas exit, the outlet nozzle is flared inside to reduce exit nozzle turbulence, thereby further reducing gas pressure drop by a significant amount.

The heating surface coil is circular in shape, made up of twenty eight flat spirally wound coils. These coils are stacked vertically. A baffle is placed across the bottom, leaving only the 42-in. gas opening. Another baffle is placed clear across the top of the coil, forcing the gas to flow radially through the coil spaces. The spiral shape of the coil elements allows for differential expansion of each coil element with respect to adjacent elements. This feature is the greatest single factor contributing to rugged reliability and its ability to operate in a dry condition for extended periods without thermal stress damage.

Extended finned tubing is used in the heating surface assembly. The tube size selected was based largely on practical rather than theoretical

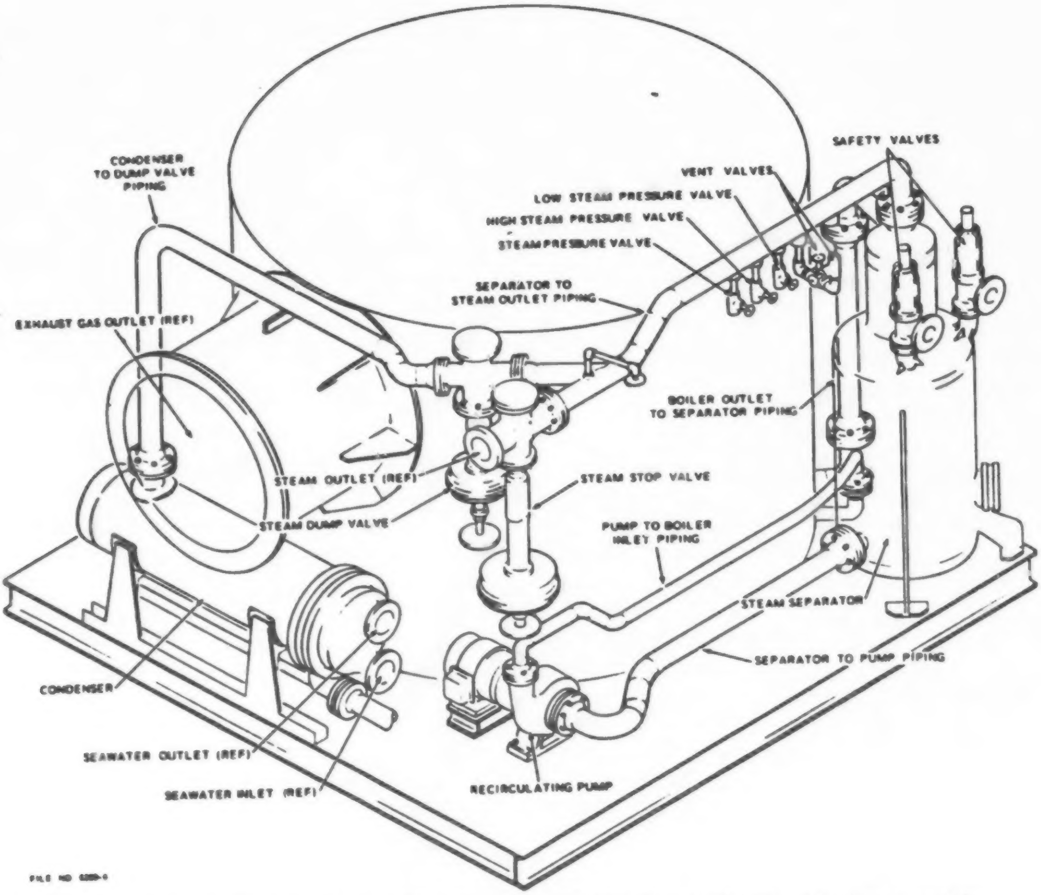
CONDENSER TO DUMP VALVE PIPING
SAFETY VALVES
VENT VALVES
LOW STEAM PRESSURE VALVE
HIGH STEAM PRESSURE VALVE
STEAM PRESSURE VALVE
SEPARATOR TO STEAM OUTLET PIPING
EXHAUST GAS OUTLET (REF)
BOILER OUTLET TO SEPARATOR PIPING
STEAM OUTLET (REF)
STEAM STOP VALVE
STEAM DUMP VALVE
PUMP TO BOILER INLET PIPING
STEAM SEPARATOR
SEPARATOR TO PUMP PIPING
CONDENSER
SEAWATER OUTLET (REF)
SEAWATER INLET (REF)
RECIRCULATING PUMP
FILE NO

Fig. 2 Heat recovery boiler arrangement of water circulating and steam system

considerations. It is true that smaller diameter tubes result in a higher gas-side and water-side heat transfer coefficients. However, the gas-side coefficient governs the overall coefficient of heat transfer and, therefore, must receive the most consideration. For example, the gas-side coefficient varies inversely as $D^{0.53}$. Therefore, for a given case of uniform gas temperature and mass flow, the ratio of coefficients of $\frac{1}{2}$-in.-O.D. tubes and 1-in.-O.D. tubes is 1.444. What this means is that almost 31 percent less surface is needed when $\frac{1}{2}$-in.-O.D. tubes are used than when 1-in.-O.D. tubes are used. If we were dealing with an organic fluid or extremely high quality water having no hardness, no solids whatsoever, this could be taken advantage of. From a practical standpoint the tube size selected is the smallest diameter tube which can pass a pneumatic tube reamer. This was done so that in the event untreated water is used for an emergency condition, or the condenser leaks seawater into the condensate, the tubes could be mechanically cleaned if chemical cleaning is impossible or the tubes become plugged to the point where chemicals cannot be introduced. The tube size used is $1\frac{1}{4}$-in.-O.D.

The fin configuration selection was on the basis of selecting a fin which would give good heat transfer, low pressure drop, and minimum fouling possibility. The segmented fin was selected because it does provide high turbulence and consequently high heat transfer rates. The segmented configuration promotes flow across the tube as well as parallel to the tube, thus making the use of soot blowing jets more effective. Fig. 4 shows a typical finned tube. Fig. 5 shows the circular coil assembly including the coil base, coil cover, and inlet and outlet manifold.

The effective area of the finned surfaces should be as large as possible in relation to the tube inside effective area, and based on practical considerations of gas pressure drop and fouling. If plain tubes were used, numerous closely packed passes would be required resulting in a larger

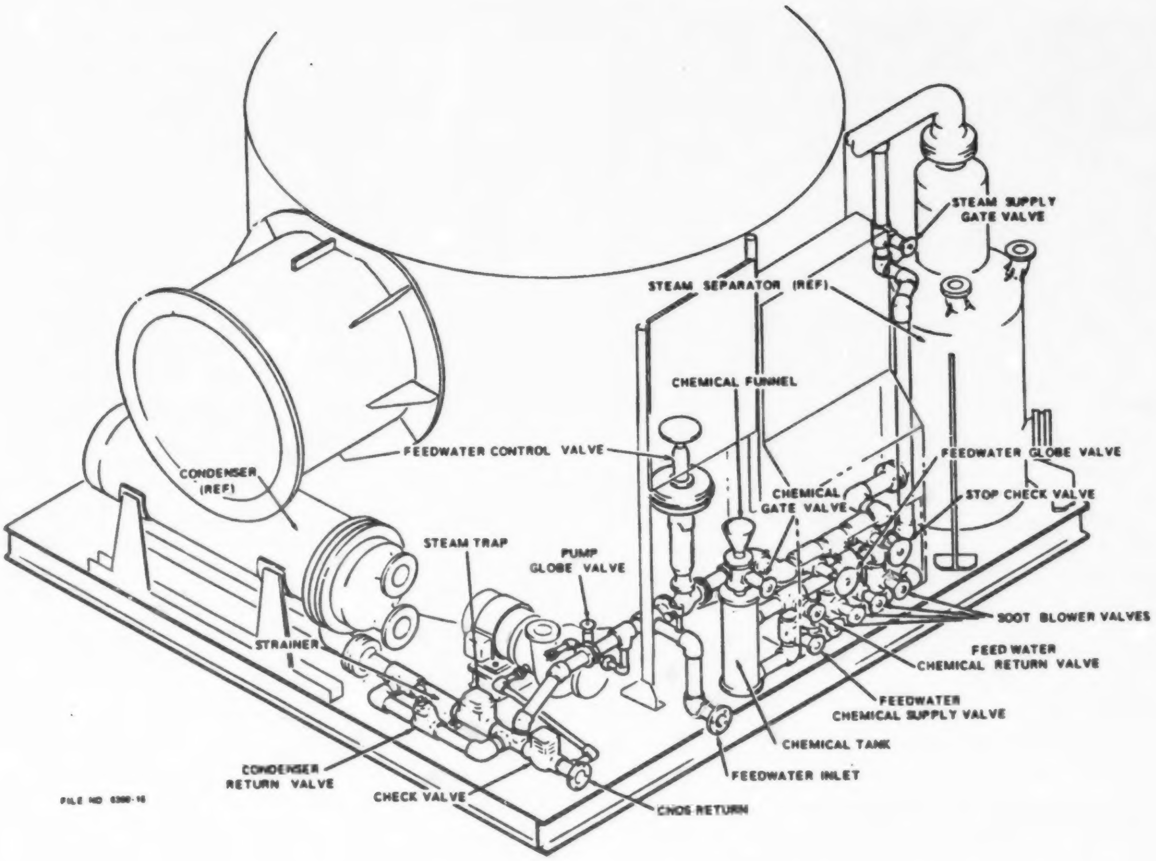
STEAM SUPPLY GATE VALVE
STEAM SEPARATOR (REF)
CHEMICAL FUNNEL
FEEDWATER CONTROL VALVE
FEEDWATER GLOBE VALVE
CONDENSER (REF)
CHEMICAL GATE VALVE
STOP CHECK VALVE
STEAM TRAP
PUMP GLOBE VALVE
SOOT BLOWER VALVES
STRAINER
FEED WATER CHEMICAL RETURN VALVE
FEEDWATER CHEMICAL SUPPLY VALVE
CHEMICAL TANK
CONDENSER RETURN VALVE
FEEDWATER INLET
CHECK VALVE
CNDS RETURN

Fig. 3 Heat recovery boiler arrangement of feedwater inlet and condensate return system

core, more gas passes, higher gas pressure drop, and greater susceptibility to fouling. As stated in the foregoing, the reasons for the use of segmented fins are higher coefficients and more cleanability. It was not considered safe to use a fin density of more than 10 fins/in. Accordingly, the fin selection was 10 fins/in., ½-in. fin height, 0.025-in. fin thickness.

The dry running requirement at 1035 F maximum temperature actually determined the tube and fin material. The tubes are SA209 Grade T-1 Carbon-moly and the fins are 11 percent chrome steel.

CONTROLS

The controls are pneumatically operated. Steam pressure is controlled by bypassing excess steam to a dump condenser. This requires that the boiler produces at its maximum rate depending on turbo-generator load which governs exhaust temperature. Any excess steam not required by the ship's services is bypassed by means of a pneumatically controlled valve to the condenser.

Water level control is accomplished by a Barton differential level controller, producing a pneumatic signal to a control valve which controls flow of feedwater to the steam separator vessel. The use of a comparatively small vertical vessel rather than a horizontal drum makes the system less affected by pitch and roll.

Recirculation rate is constant and not controlled except for a limiting orifice which produces the head required to limit recirculated water flow to the maximum set by the boiler's maximum evaporating capability.

The pneumatically operated emergency stop valve shuts off the flow of steam to the ship's system in the event of:

- Abnormally high steam pressure
- Abnormally low steam pressure
- Loss of control air
- Low water level

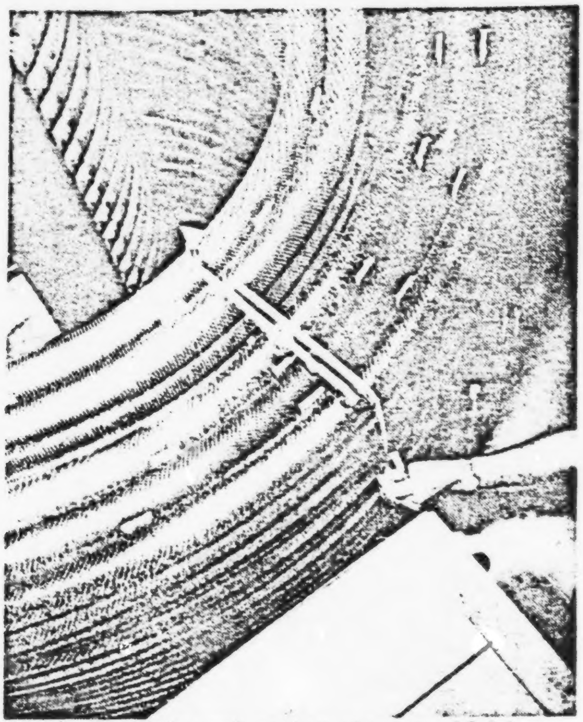

Fig. 4 Fin tube and center steam manifold

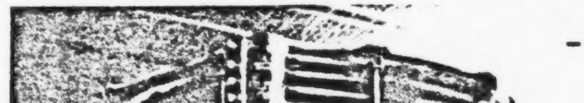

Fig. 5 Heat recovery coil assembly with feedwater inlet manifold and soot blower manifold

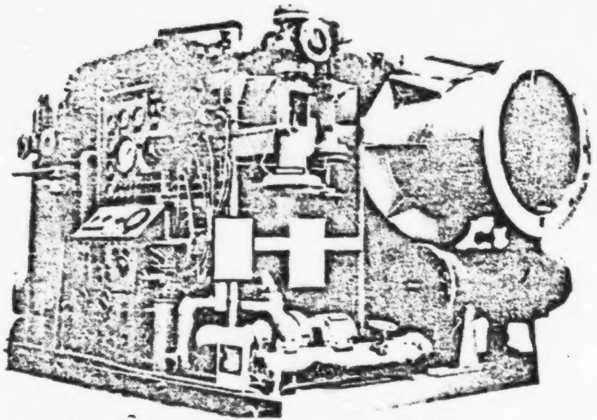

Fig. 6 Heat recovery boiler assembly

- High water level.

PHYSICAL ARRANGEMENT

Fig. 6 illustrates the completed boiler assembly. This assembly includes all of the accessories and controls required to operate the boiler with the exception of the boiler feedwater pump which is remotely located. The control panel is in the front or operating side. Two arrangements are being manufactured, one having the exhaust duct on the right and one having it on the left side. The condenser is mounted directly under the exhaust duct. The recirculation pump is on the same side but forward. The steam separator-reservoir assembly are on the opposite side toward the front of the base. Control valves are grouped together near the front and adjacent to the control panel.

In spite of the fact that critical components are located in seemingly accessible places, actually the space allotment and bulkhead arrangement on the ship makes accessibility to some components difficult.

QUALIFICATION PROGRAM

The Qualification Program consisted of a series of tests. The shock and vibration tests were not performance oriented and, thus, would not be covered in this report.

The Performance Qualification Test was conducted at Conseco's test facility. Two 30,000,000-Btu oil-fired combustors and a 50-lb/sec, 200-hp blower and associated controls and instruments were used to simulate the gas turbine exhaust. Feedwater tank and pump, along with condenser cooling water, were shipboard simulated.

By varying the amount of fuel to the combustor and the amount of ambient air to the blower, the boiler was fired at various exhaust conditions of which performance curves were generated.

Some of the other tests that were performed included:

- Maximum boiler performance longevity test
- Sudden application and removal of steam load
- Emergency stop
- Manual operation of the boiler
- Operation of the boiler under casualty conditions
- Boiler response time under sudden application of exhaust gas
- Dry running at maximum inlet exhaust gas conditions
- Steam quality test
- Boiler condenser maximum capacity test
- Boiler blowdown test
- Soot blower and gas side cleaning test.

The test results point out some interesting facts. Even under prolong dry running condition of 12 hr at 1050 F, the boiler tubes did not exhibit any overstressing. The boiler response time under instant start was surprising. It took 4 min. at operating conditions, i.e., 650 F and 32-lb/sec flow for the boiler to generate 100-psi steam, consequently, 2½ min. at 1000 F.

Using a calorimeter, the steam quality was measured at 99.7 percent during boiler operating conditions. The boiler controls and liquid level were quick to react under sudden application and removal of steam loads.

RECOMMENDATIONS FOR FUTURE NAVAL HEAT RECOVERY UNITS

Basic Sizing Considerations

The boiler design criteria were based on turbine maximum exhaust temperature of 1035 F. As such, the boiler coil, manifold, circulating pump, and system were designed for a steaming rate compatible with this temperature. However, in actual operation on the ship, the boiler gas inlet temperature was never in excess of 840 F, and 90 percent of the time it is between 650 and 750 F. This philosophy results in an over-sized boiler in terms of weight and physical dimensions.

It is recommended that the boiler sizing criteria be established by the normal operating conditions in conformance with the actual electrical loads expected, regardless of the turbine maximum operating capabilities. After the boiler is sized for that particular requirement, abnormal conditions such as high ambient conditions, and battle conditions should be calculated for information purposes and to determine system safety conditions. Correction for additional steam capacity to meet a specific requirement should then be made.

Physical Arrangement of Boiler Components

Utilization of the forced recirculation cycle makes possible the arrangement and location of components so that accessibility for maintenance and replacement of parts are minimized.

If the space problem is such that a complete integral boiler would cause problems in accessibility, the components can be dispersed so that the heating surface coil assembly can be located in the uptakes, the circulating pump and steam separator assembly in the machinery space and the control panel in the control room. The interfaces between the components would be reduced to conventional pipe runs and measurement-type control lines.

In almost all cases, the coil assembly can be easily fitted into the gas turbine exhaust duct, with the inlet manifold built into the duct casing wall. The manifold will have an inspection plug opposite each tube circuit for easy inspection and/or clean-out when required.

The evaporator tubes can have any orientation or length and do not have to be vertical and of a definite length as is the case with a natural circulation boiler.

Controls

Controls should be simplified to the greatest extent possible. Pneumatic liquid level controls should be of the positive indicating type. Water-filled legs should only be used whenever there is sufficient static head between the liquid level and the instrument, such as in common two-drum boilers. In cases where a small vertical separator is used, sealed liquid filled legs with bellows are recommended.

Since the boiler should be designed for dry running, controls, such as high and low level switches, should be eliminated. A simple device to indicate no steam flow, no water flow, or a drop in boiler pressure should be used to indicate a boiler malfunction.

Pressure, temperature, and flow transducers could be used in conjunction with an electronic control center to take over all control and alarm functions.

Circulating Pump

Standard Navy packed pumps have been found to be unsuitable. The combination of high water temperature and the boiler water chemicals caused excess packing leakage. Mechanical water cooled seals of the carbon and ceramic type have been found to be the most suitable.

Dry Running Capability

The boiler should be designed for dry running capability on a continuous basis at gas temperatures up to at least 900 F for the following reasons:

1 Since boiler's internal damage as a result of dry running is eliminated, controls can be simplified.

2 Cleaning of the gas-side can be accomplished with a high degree of effectiveness.

3 Gas turbine can operate in the event of boiler failure.

Safeguards should be provided to eliminate admittance of cold water to a hot boiler to avoid the effects of thermal stresses.

Gas-Side Cleaning

Gas-side cleaning was originally the most worrysome problem. Four stationary soot blowers have been positioned inside the coil with limited effectiveness (Fig. 7). Six or possibly eight would have been better. If height constraints had permitted, a single automatic rotating soot blower would have been positioned in the coil center which would have had virtually 100 percent coverage.

After about 10,000 hr of running time, the following was noticed: During the first 100 hr, a gas pressure drop increase of about 0.75 in. was noted, after which the pressure drop remained constant with periodic soot blowing. This rapid increase could also have been due to repeated start-ups in the first 100 hr, each start-up discharging unburned fuel into the boiler which condensed onto the tubes and provided a base for soot accumulation. For the Iranian DD program, the addition of four more soot blower circuits is proposed to give more effective coverage.

During extensive testing at Conseco's test facility for gas-side fouling, it was concluded that the most efficient gas-side cleaning was by dry running. At 750 F exhaust gas temperature, it required about 25 min. to reduce the back pressure to its original "as new" condition. Similarly, at 850 F it required 15 min., and at 925 F it required 5 to 10 min.

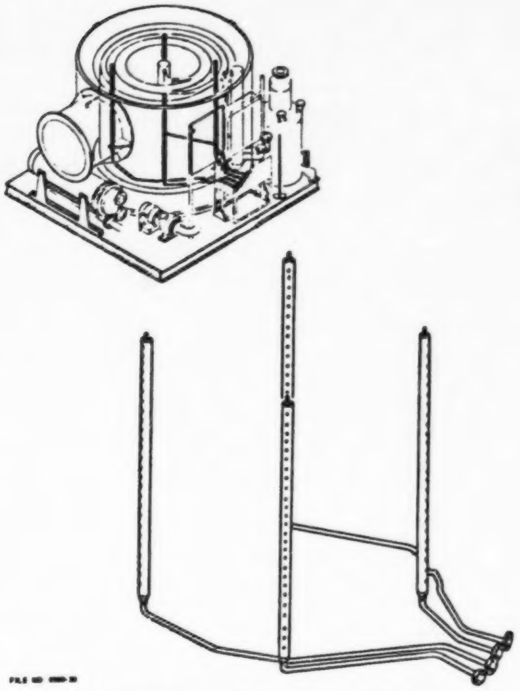

Fig. 7 Heat recovery boiler soot blower arrangement

Another means for cleaning the gas-side, which is about 90 percent effective, is the periodic injection of Gamlenite 12 dry compound into the hot gas stream into the boiler. It required about four charges of one pound per charge while the boiler was in the operating mode to reduce the back pressure by 90 percent. However, once the surfaces are in clean condition, it is estimated that one-half pound charge per day will maintain them in reasonably clean condition.

Unfortunately, present conditions are such that a load bank would be required in order to increase the exhaust gas temperature above 700 F. Perhaps the cleaning operation can be performed when electrical load is such as to produce an exhaust temperature in excess of 700 F.

It is highly recommended that any future installation incorporate dry running conditions to efficiently clean any fouling deposits on the gas-side.

The use of Gamlenite 12 would also have been more effective if it had been applied when the boiler was in the dry running condition because tube surface temperature would have been much higher than in the steaming mode, even if gas temperature is as low as 650 F. Tube and fin temperatures never reach 650 F when in the steaming mode, even when gas temperature is as

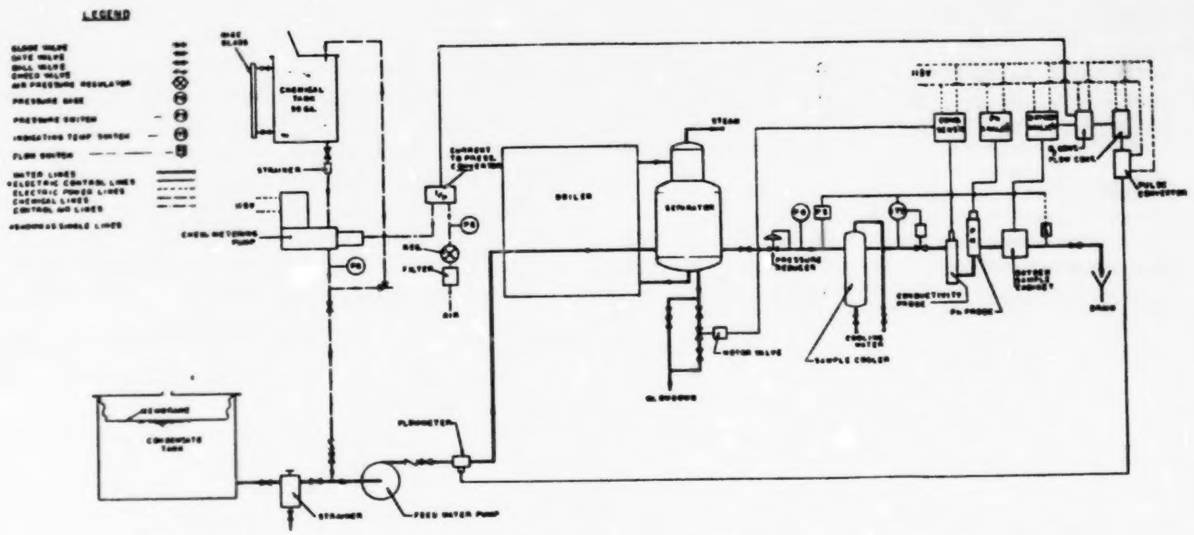
LEGEND
GLOBE VALVE
GATE VALVE
BALL VALVE
CHECK VALVE
AIR PRESSURE REGULATOR
PRESSURE GAGE
PRESSURE SWITCH
INDICATING TEMP SWITCH
FLOW SWITCH
WATER LINES
ELECTRIC CONTROL LINES
ELECTRIC POWER LINES
CHEMICAL LINES
CONTROL AIR LINES
SIGHT GLASS
CHEMICAL TANK
STRAINER
CHEM. METERING PUMP
CURRENT TO PRESS. CONVERTER
REG.
FILTER
AIR
BOILER
STEAM
SEPARATOR
PRESSURE SENSOR
MOTOR VALVE
BLOWDOWN
SAMPLE COOLER
COOLING WATER
CONDUCTIVITY PROBE
PH PROBE
OXYGEN SAMPLE CABINET
FLOW CONT.
PULSE CONVERTER
DRAIN
MEMBRANE
CONDENSATE TANK
FLOWMETER
STRAINER
FEED WATER PUMP

Fig. 8 Heat recovery boiler liquid chemical feedwater treatment schematic diagram

high as 950 F. It is estimated that tube and fin temperatures are in the 390 to 450 F range under the foregoing conditions.

FEEDWATER TREATMENT

The feedwater system on the present DD ships does not include a deaerating feedwater heater. Accordingly, since the system is open, including an atmospheric condensate return tank, the water contains dissolved oxygen almost up to saturation. The condensate tank operates at 130 to 150 F which is not high enough to drive off all of the oxygen.

Feedwater treatment consists of periodic dosing of the system with Navy compound through the small chemical tank mounted on the boiler. A portion of the feedwater is diverted through this tank, carrying with it the chemical contents of the tank.

There are difficulties with this system caused by incomplete solution of the chemicals, the addition of doses instead of constant uniform feeding, and the fact that the compound does not contain oxygen scavenging materials. The limited water capacity of the forced recirculation system, the short time period for complete solution, and other factors make this system virtually ineffective as a water treatment means for a high performance forced recirculation system.

Since makeup comes from the ship's evaporator, the water contains but little hardness, a small amount of chloride carry-over, is low in pH (6-6.5), and high in dissolved oxygen. Therefore, the primary concern is to remove oxygen, raise pH to 8.5 to 9, and treat what little hardness exists.

It is proposed that:

1 Future ships be equipped with deaerating feedwater heaters for oxygen removal or that a chemical feed system constantly proportioning liquid chemicals for oxygen scavenging
2 Chemical metering for pH control by automatic means
3 Chemical metering for hardness elimination (if present).

All of the foregoing can be accomplished by the metering of one chemical into the feedwater line at the feedwater pump. The chemical will be in liquid form and will be quickly absorbed into the water stream. Control means can be provided for oxygen and pH control.

4 Automatic blowdown based on conductivity of the water in the separator vessel
5 Flotation of a polyethylene membrane on the condensate water level to reduce the interface surface of water to air.

Two available chemicals are: (a) Drew Chemical Co.; AGK 100 and (b) Nalco Chemical, Nalpak 8240.

It is estimated the required injection

rate based on an average steaming rate of 7000 lb/hr is 0.022 gal/hr. A schematic diagram is illustrated in Fig. 8.

CONCLUSION

The foregoing shows that it is entirely feasible to produce successful exhaust heat recovery equipment for use on Naval vessels as well as on commercial ships. The experience gained in the DD-963 application, which is, in itself, a reliably successful system, will lead to even more efficient means for recovering heat from gas turbine exhaust in the future. The same heat recovery technology can be applied to marine combined cycle applications, and organic bottoming cycles which can play an important role in marine propulsion development.

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DD-963 Class Waste Heat Recovery System Experience

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Philadelphia, Pa.

The DD-963 Class ships are the first U.S. Navy vessels to utilize a waste heat recovery system on a gas turbine engine. This paper will present the experience gained from the three years of shipboard operation with the system. The experience will be used to develop areas for consideration that can improve the probability of success in future system procurements. The areas to be considered are: (a) the need for definitive military specifications, (b) the need to test at Navy laboratories and (c) the need to test complete systems under simulated shipboard conditions.

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T. E. GRAF **J. E. NAGENGAST**

ABSTRACT

The DD-963 Class ships are the first U.S. Navy vessels to utilize a waste heat recovery system on a gas turbine engine. This paper will present the experience gained from the 3 years of shipboard operation with the system. The experience will be used to develop areas for consideration that can improve the probability of success in future system procurements. The areas to be considered are (1) the need for definitive military specifications; (2) the need to test at Navy laboratories and (3) the need to test complete systems under simulated shipboard conditions.

INTRODUCTION

The first US Navy DD-963 Class Ship, USS SPRUANCE, was commissioned in October 1975. The other ships of the class have followed closely behind such that 3 years later more than half of the 30 ship class have been commissioned. These ships are the first in the U.S. Navy to utilize a Waste Heat Recovery System (WHRS) on gas turbine engines. To date two papers, (1, 2) have been published by the boiler manufacturer, Conseco Inc. on the subject of DD-963 waste heat boiler design and a proposed future design. It is the purpose of this paper to present and discuss the experience gained from the users viewpoint, the U.S. Navy, after 3 years of shipboard operation. Some problems that have occurred and the corrective action taken will be discussed. Lastly, consideration is given to ways in which these problems can be avoided in future WHRS acquisitions for U. S. Navy.

DESCRIPTION

The DD-963 Class ship has 3 waste heat recovery boilers installed in the exhaust duct of each of the 3 ship's service gas turbine generators (SSG). The generators are rated at 2000 kw each and driven by Allison 501-K gas turbines.

Table I gives the design parameters of the boiler at rated condition.

TABLE 1. DESIGN PARAMETERS OF DD-963 WASTE HEAT RECOVERY BOILER

Nomenclature	*Value at Rated Point*
Steam Output Capacity	*0.88 kg/s(7000 lb/hr)*
Steam Operating Pressure	*689 kPa(100 psig)*
Feedwater Temperature	*333 K (140°F)*
Turbine Exhaust Gas Temperature	*616 K (650°F)*
Boiler Exhaust Gas Temperature	*478 K (400°F)*
Gas Mass Flow Rate	*15.5 kg/s (34.2 lb/sec)*

The boiler is of the unfired, forced circulation type with a circulation ratio (water to steam) of approximately 10 to 1. The amount of steam generated is directly dependent on the amount and temperature of the gas turbine exhaust gas flow. The steam flow in excess of that required by the ship's services is diverted through a control condenser where it is condensed and returned to the condensate system. The steam is distributed to the various services throughout the ship as shown on Figure 1. The steam is returned as condensate from the system via a drain cooler to a feedwater and drain

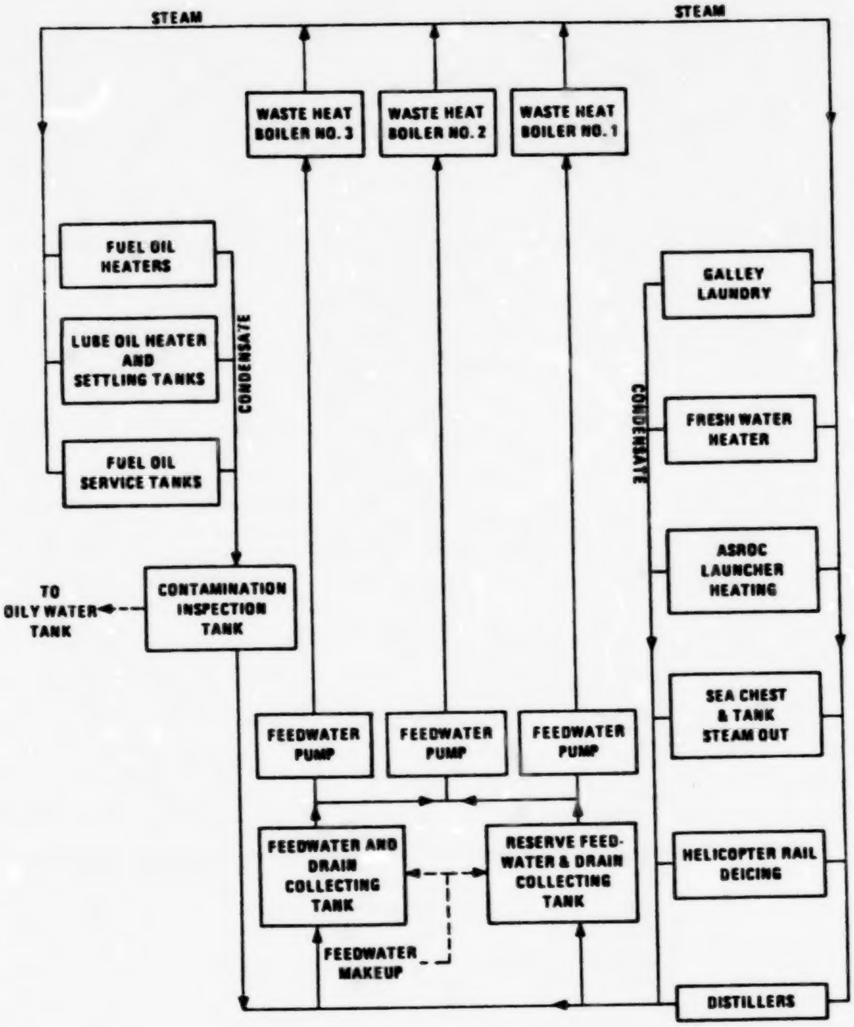
STEAM
STEAM
WASTE HEAT BOILER NO. 3
WASTE HEAT BOILER NO. 2
WASTE HEAT BOILER NO. 1
FUEL OIL HEATERS
LUBE OIL HEATER AND SETTLING TANKS
FUEL OIL SERVICE TANKS
CONDENSATE
GALLEY LAUNDRY
FRESH WATER HEATER
ASROC LAUNCHER HEATING
SEA CHEST & TANK STEAM OUT
HELICOPTER RAIL DEICING
CONDENSATE
TO OILY WATER TANK
CONTAMINATION INSPECTION TANK
FEEDWATER PUMP
FEEDWATER PUMP
FEEDWATER PUMP
FEEDWATER AND DRAIN COLLECTING TANK
RESERVE FEED-WATER & DRAIN COLLECTING TANK
FEEDWATER MAKEUP
DISTILLERS

Figure 1. DD-963 Class Ship System

collecting tank. The feedwater pump takes suction from the tank and supplies feedwater to the separator. Makeup water is automatically supplied to the drain collecting tank from the distilling plant. The boiler system is shown on Figure 2.

The boiler is capable of automatic starting and stopping in conjunction with SSG. Once the feedwater and circulating pumps are started, the boiler is automatically operated as the gas turbine generator set is brought on-the-line and up to speed. The boilers are generally in operation whenever the engine operates since the gas flow is not diverted around the boiler. The boilers have been designed for "dry firing". This is gas flow through the boiler but on steam generation. Normal mode of operation is two boilers on-the-line during underway periods and one boiler on-the-line when in-port.

DISCUSSION OF DESIGN AND ASSOCIATED PROBLEMS

The following is a discussion of some of the problems that have surfaced with the waste heat recovery system during the first 3 years of shipboard operation. Included are the corrective actions taken to correct the problems and the results that have occurred.

Gas Flow Differential Pressure

The design pressure drop across the boiler coil from inlet to exhaust duct connection was specified at 1.1 kPa (4.5 in.H_2O) at rated exhaust gas flow. During the initial boiler qualification test performed by the manufacturer at his facility, the pressure drop was measured at 1.2 kPa (4.9 in.H_2O) at the beginning and increased to 1.5-1.6 kPa (6-6.5 in.H_2O) during the 100 hour test. When the unit was installed on 2.5 kPa (10.0 in.H_2O) with the boiler in a clean condition. The qualification testing was conducted at the manufacturer's facility using a combustion chamber rather than a gas turbine engine to produce the gas flow. This in a 100% increase in the pressure drop across the boiler. A design change was developed by the manufacturer consisting of air diverting blades installed in the inlet duct of the boiler. The blades direct the gas flow radially out through the coils and eliminate the gas turbulence in the inlet portion of the coil. Three of the blades are bolted to brackets on the outlet header and the fourth blade is welded to the top of the boiler (See Figure 3). The air diverter blades

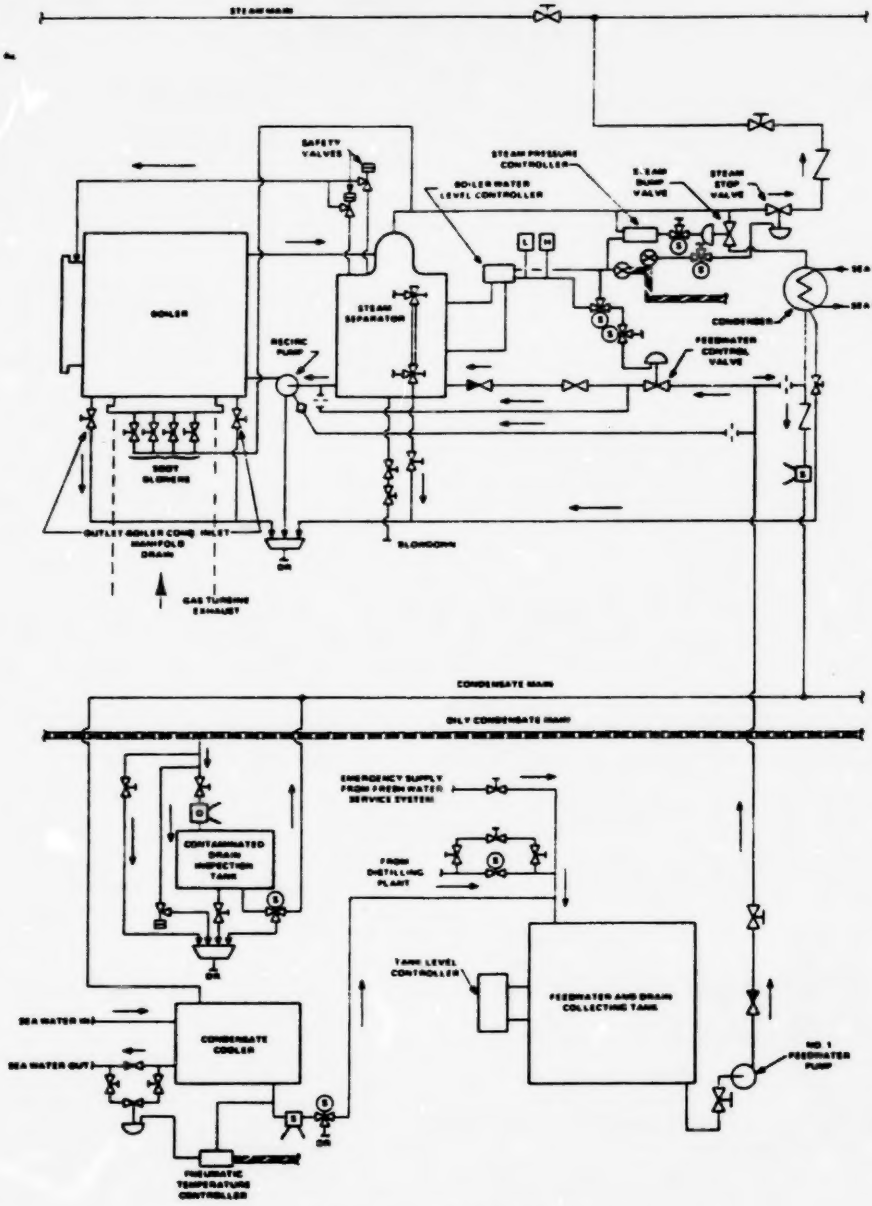
STEAM MAIN
SAFETY VALVES
STEAM PRESSURE CONTROLLER
BOILER WATER LEVEL CONTROLLER
STEAM DUMP VALVE
STEAM STOP VALVE
L
H
S
SEA WATER IN
SEA WATER OUT
BOILER
STEAM SEPARATOR
CONDENSER
RECIRC PUMP
FEEDWATER CONTROL VALVE
SOOT BLOWERS
OUTLET-BOILER CONN-INLET
MANIFOLD DRAIN
DR
BLOWDOWN
GAS TURBINE EXHAUST
CONDENSATE MAIN
OILY CONDENSATE MAIN
EMERGENCY SUPPLY FROM FRESH WATER SERVICE SYSTEM
CONTAMINATED DRAIN INSPECTION TANK
FROM DISTILLING PLANT
TANK LEVEL CONTROLLER
FEEDWATER AND DRAIN COLLECTING TANK
SEA WATER IN
CONDENSATE COOLER
SEA WATER OUT
NO 1 FEEDWATER PUMP
PNEUMATIC TEMPERATURE CONTROLLER

Figure 2. DD-963 Waste Heat Boiler System

reduced the pressure drop by 0.5 kPa (2.0 in.H_2O). This modification has been incorporated in all of the boilers.

Additional changes have been proposed by the manufacturer to reduce the gas pressure drop in the area where the exhaust duct connects to the boiler casing. The first change was to cut a 45 degree flare opening between casing inner wall and the exhaust duct. This increased the opening of the exhaust duct from 0.9 m (36 inch) to 1.0 m (40 inch) diameter and provided a smooth transition from the casing into the exhaust duct. The second change was to install diverting vanes from the coils into the exhaust duct. Testing by the manufacturer showed a reduction of 0.5 kPa (2.2 in.H_2O) in the overall pressure drop from the enlarged flare opening and a reduction of 0.2 kPa (0.8 in.H_2O) from the diverting vanes. These two changes have been incorporated on the later boilers not yet installed but have been judged to be too costly to be backfitted on the boilers that were installed

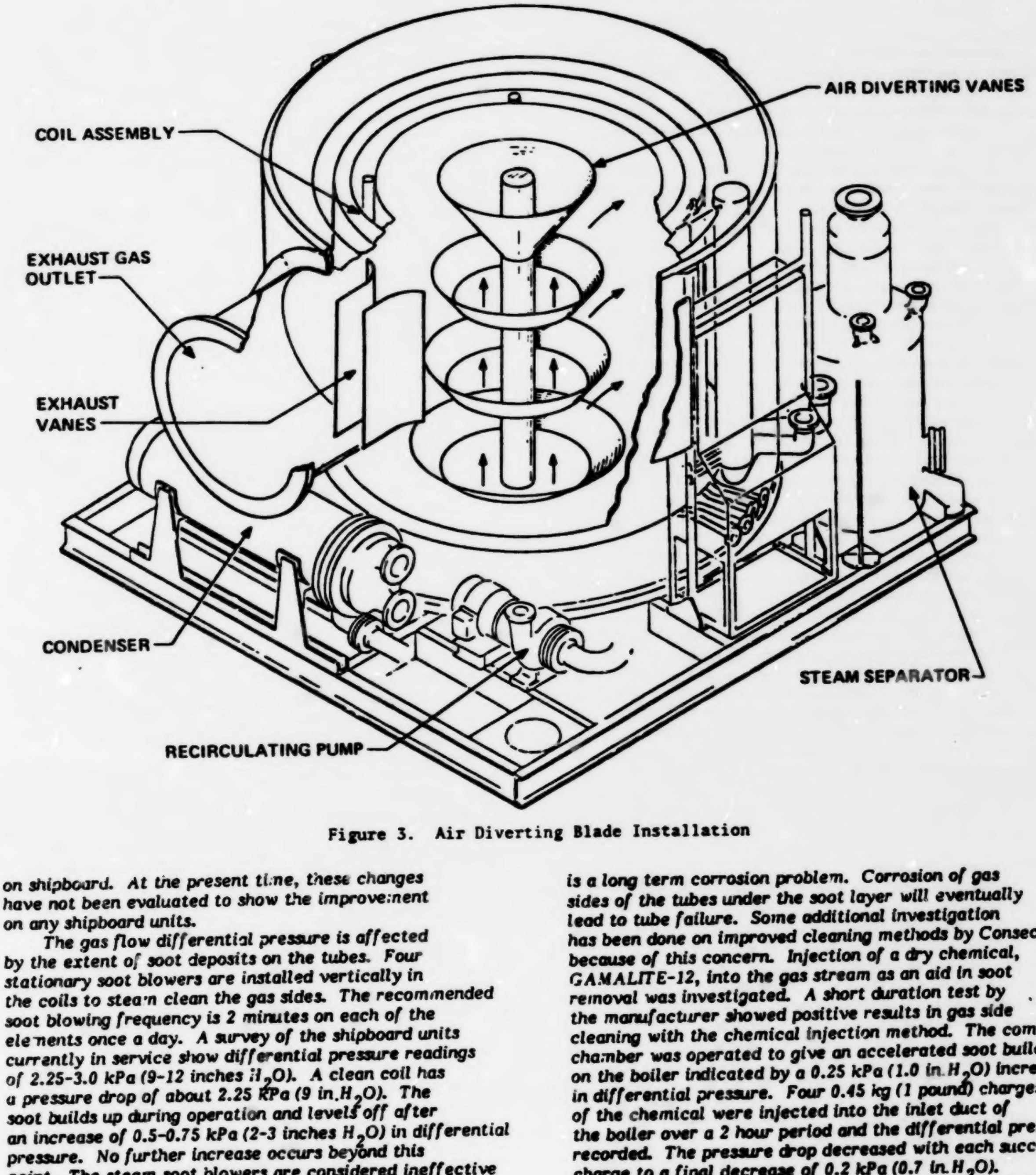

Figure 3. Air Diverting Blade Installation

on shipboard. At the present time, these changes have not been evaluated to show the improvement on any shipboard units.

The gas flow differential pressure is affected by the extent of soot deposits on the tubes. Four stationary soot blowers are installed vertically in the coils to steam clean the gas sides. The recommended soot blowing frequency is 2 minutes on each of the elements once a day. A survey of the shipboard units currently in service show differential pressure readings of 2.25-3.0 kPa (9-12 inches H_2O). A clean coil has a pressure drop of about 2.25 KPa (9 in.H_2O). The soot builds up during operation and levels off after an increase of 0.5-0.75 kPa (2-3 inches H_2O) in differential pressure. No further increase occurs beyond this point. The steam soot blowers are considered ineffective in removing the accumulated soot because of the small boiler coil area covered by each element. The results of shipboard survey showed little change in the differential pressure after soot blowing. An alternate method proposed by Conseco was to install an automatic rotating soot blower in the coil center. This unit would have provided 100% coverage throughout the tube coils. However, the height constraint on the ship would not permit the use of this type of soot blower.

The primary concern with the soot formation and the inability of soot blowers to remove the soot is a long term corrosion problem. Corrosion of gas sides of the tubes under the soot layer will eventually lead to tube failure. Some additional investigation has been done on improved cleaning methods by Conseco because of this concern. Injection of a dry chemical, GAMALITE-12, into the gas stream as an aid in soot removal was investigated. A short duration test by the manufacturer showed positive results in gas side cleaning with the chemical injection method. The combustion chamber was operated to give an accelerated soot buildup on the boiler indicated by a 0.25 kPa (1.0 in.H_2O) increase in differential pressure. Four 0.45 kg (1 pound) charges of the chemical were injected into the inlet duct of the boiler over a 2 hour period and the differential pressure recorded. The pressure drop decreased with each successive charge to a final decrease of 0.2 kPa (0.7 in.H_2O). While this method shows promise, it must be considered as just a laboratory demonstration. Additional development work is required to make this method into an effective cleaning method for shipboard use. Secondly, this method would require the installation of a chemical injection system on each boiler inlet duct. Supply support onboard the ship and at the depot level for the dry chemical would be necessary.

Sustained high temperature, dry operation of the boiler to remove soot deposits has long been considered as a potential method. The Navy has recently sponsored *some investigation efforts in gas side fouling and cleaning. The results of this study indicate that tube metal temperature of approximately 700-755 K (800-900 F) is required to effectively remove the soot deposits. This can be achieved by running hot gases at a temperature of 727-783 K (850-950 F) through a dry (without water) boiler.*

To utilize dry operation of a boiler as a cleaning method is not necessarily an easy method on shipboard. On the DD-963 class ships, the normal underway operating mode is two generator sets on-the-line with each producing about 1000 kw. This gives an engine exhaust gas temperature of 589-644 K (600-700 F). The operation of one generator set would load the engine enough to give the needed 727-783 K (850-950 F) gas temperature for effective cleaning. However, the forces afloat are reluctant to rely on one generator set for ship power when underway. Secondly, the production of steam has been stopped during the cleaning period, which would necessitate shutting down the distilling plants and other steam services. It is possible to use the dry operation method when the ship is in-port, but a load bank would be required to increase the electrical load on the generator and achieve the necessary gas temperature. In addition, the boiler must be allowed to cool down for a 4-6 hour period after each cleaning before it can be returned to service.

There is only minimal impact on gas turbine engine performance as a result of the increase in differential pressure. An increase in differential pressure across the gas side of the boiler results in an increased back pressure on the gas turbine engine. For the Allison engine an increase of 1.25 kPa (5 inch H_2O) causes a 1 percent increase in fuel consumption. The effect of the increase in back pressure on operation is minimal, since the decrease in engine and generator power has no limiting effect on the overall power capacity of the ship. Likewise, the loss of steam generating capacity due to the soot formation on the boiler coil is small in comparison to the total steam capacity. The primary concern of the soot formation on the boiler is a long term corrosion problem.

Waterside Corrosion

The first problem with the corrosion of the boiler was discovered in May 1975 during construction of the DD-964. The boilers sat idle for approximately 1 1/2 years between construction and test on the ship. Following acid cleaning of the unit, a hydrostatic test revealed two tubes had failed on No. 2 unit. A detailed analysis was made of these tubes by an independent test laboratory. A general corrosion condition was found on the inside surfaces of the boiler tubes. The perforations of the tube wall was caused by pitting corrosion under stagnant conditions with failed tubes partially filled with water (See Photographs in Figure 4). The boiler had been preserved with a soluable oil and water mixture,by the manufacturer which did not provide adequate protection during the long time period before operation. The specification for preservation has been changed and utilizes a true corrosion preventive solution, CESCO Film-Guard 140. There has been no recurrence of this problem.

A second corrosion problem was found on the DD-963 when the boilers were put into service. The complete boiler, steam and water system were contaminated with solid particulate matter, which was determined to be iron oxide. The condition was such that the shipboard personnel had difficulty in conducting the boiler water chemical tests to maintain proper treatment. The boiler water sample cooler, air ejector nozzles, and other components were found to be plugged solid with debris. The boiler and attached piping located on the baseplate are fabricated from ferrous material and were found to be the source of iron oxide. The problem was not one of improper material selection but a result of the inadequate preservation of the boilers between construction and initial operation. This was compounded by an ineffective cleaning/flushing method used by the shipbuilder. An improved boiler chemical cleaning procedure using tri-sodium phosphate to remove the preservative and inhibited hydrocloric acid to dissolve the iron oxide has been instituted to ensure a clean unit at initial light-off.

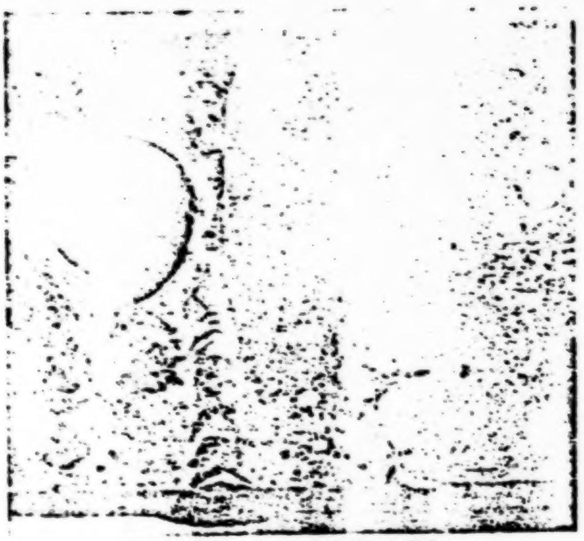

Figure 4. Failed Boiler Tube

Beyond this initial corrosion problem, there existed a continuing problem with rust due to inadequate boiler lay-up. An inspection of the USS SPRUANCE boilers approximately 6 months after being put into service showed a continued rust problem in samples of water taken from the steam separator. For less than a 30 day shutdown, documentation called for securing the boiler and keeping it in a standby mode partially full of water. For shutdown greater than 30 days, the boiler was to be kept completely full of water. Neither of these methods are satisfactory, because whenever an air-water interface exists, the boiler will be subjected to oxygen corrosion. Air will enter a partially filled boiler through small leaks in the system when it is secured. With the long term lay-up, it was found that the water in the boiler would drain down to the feedwater tank and the boiler could not be kept full of water. Both methods caused an air-water condition to exist in the boiler. A short term lay-up method has been developed using a steam blanket on the idle boiler and the addition of a check valve to prevent the loss of water. The steam blanket eliminates the air in the boiler by bleeding steam back into the boiler from the ship's steam system. The steam blanket lay-up is a standard method of boiler lay-up used by the Navy for periods of less than 60 days. A dry lay-up method has been developed for lay-up longer than 60 days. The boiler is shutdown and drained of all water. The boiler is then run dry at 533 K (500 F) for 4 minutes to give a complete dry-out of the watersides. Following cooldown, the boiler is isolated by closing all of the system valves. These improved methods will eliminate the inadequate lay-up problem.

A boiler corrosion problem also existed that is *related to operation. The feedwater tank is open to the atmosphere. The temperature of feedwater in the tank is maintained at 333 K (140 F) and is capable of absorbing dissolved oxygen as high as 6 ppm. Under these conditions, the ferrous material of the boiler proper is susceptible to oxygen attack. Several of the ships had experienced failures of feedwater piping due to oxygen corrosion. A new boilerwater/feedwater treatment system has been installed for evaluation. The treatment system includes the use of an oxygen scavenger, hydrazine, to eliminate the problem of oxygen corrosion. The results of shipboard evaluation showed evidence of satisfactory oxygen scavenging, characterized by black appearance of boiler watersides. The black appearance indicates a passive condition rather than a red active corrosion condition. The feedwater treatment change along with the preservation and lay-up procedure changes should eliminate all oxygen corrosion problem being experienced with the WHB's.*

Boilerwater Treatment

The initial treatment provided for the boilers was Navy Boiler Compound (NBC) which is added periodically to a tank and injected into the separator by diverting feedwater through the tank. The amount of chemicals used is based on sampling and testing of the water in the steam separator. The treatment is intended to minimize boiler metal corrosion and aids in the prevention of scale on boiler watersides. The sludge that forms in the boilerwater as a result of the treatment is removed by periodic blowdown of the steam separator. A problem with maintaining the amount of chemicals in the feedwater within the prescribed limits was first experienced on USS SPRUANCE. Investigation by NAVSECPHILADIV initially attributed the problem to the inexperience of the shipboard operators. The operators had no experience with water chemistry and had been given little formal training. It was also found that the feedwater was being contaminated with shorewater containing silicia during in-port periods, a common problem on naval ships. The treatment was changed to High Phosphate to enable the operators to control this contamination. After visits to several other ships, it was determined that the problem of maintaining the water treatment within the prescribed limits was not primarily due to operator inexperience but was due to chemical hideout. Chemical hideout is a condition where the treatment chemicals adhere to the boiler watersides during operation rather than staying in solution. The condition is characterized by a loss of chemicals following treatment of the boiler water, which can be interpreted as under-treatment of the boiler water. The chemicals will then return to solution when the boiler is later shutdown, which can be interpreted as over-treatment.

After the problem was identified as chemical hideout, the cause of the problem could be investigated. The conditions that can cause this type of problem are waterside deposits, tube water starvation and hot spots due to turbulent gas flow. Waterside deposits provide a means for the chemical to hideout. Generally dirty watersides can be corrected by chemical cleaning to remove the deposits. However, an inspection of the watersides following chemical cleaning to make a determination of cleaning effectiveness, is difficult because of the boiler configuration. The condition of tube water starvation is caused by inadequate water circulation through tube coils which leads to superheating of the water and an accelerated deposit buildup. Hot spots due to turbulent gas flow also causes localized overheating and an accelerated deposit buildup. Sophisticated instrumentation and testing is required to identify which tube areas may be overheating and design changes would be required to correct the problem.

An alternate solution to the problem of chemical hideout was developed by the Navy. To improve waterside treatment a continuous chemical treatment system was installed on USS SPRUANCE as a prototype for evaluation. The chemical treament solution used was Drew Chemical Corporation AGK-100. This is a composite solution intended to prevent scale and sludge deposit, and oxygen corossion. The AGK-100 was reinforced with Amerzine to provide additional oxygen scavenging because of the open feed system. The system includes a storage tank and a metered injection pump connected at the discharge of the feedwater pump. The system dosage is controlled by changing the concentration of solution in the storage tank or by varying the injection rate of pump, based on daily boiler water testing.

The results of a six month shipboard test showed satisfactory results in terms of correcting the oxygen corrosion problem, as previously discussed. In addition, the AGK-100 solution did provide better control of treatment limits and it was concluded that it is capable of providing effective water treatment for the waste heat boiler. However, chemical hideout to a lesser degree was still present in the boiler. As a result of this evaluation program, continuous chemical injection systems are being installed on all ships of the class.

Additional work needs to be done to completely eliminate the chemical hideout condition. Proper treatment in conjunction with blowdown will mimimize or eliminate deposit buildup on the watersides of the tubes. When proper treatment cannot be maintained, deposits will build up. The consequence of waterside deposits is not as serious on an unfired boiler as it would be on a fired boiler, localized overheat and tube rupture will not occur. However, in an unfired boiler localized corrosion cells are set up under tube deposits and this can cause long term failure if the tubes are not periodically cleaned. In addition, the chemical hideout leads to overtreatment of the boilers. This is a useless expenditure of expensive treatment chemicals.

Tube Failures

As previously noted, two boiler tubes had failed on USS PAUL F FOSTER (DD-964) as a result of inadequate preservation. Tube failures have occurred on USS SPRUANCE (DD-963) and USS BRISCOE (DD-977) as a result of broken gas baffles in the inlet duct of the boilers. The broken baffles caused a hole to be rubbed through the tube wall. The first occurrance on USS SPRUANCE was corrected by a redesign of the joining method and a material change. The second occurrance which was on the USS BRISCOE was with the modified design and it appears that a second look must be taken at this problem.

In mid 1978 tube failures began to occur on USS SPRUANCE (DD-963) boilers at the tube to inlet header joint. (See Figure 5). The failures were identified as circumferential cracks. As of September 1978 the following failures had occurred:

No. 1 WHB - 3 tube failures
No. 2 WHB - 2 tube failures
No. 3 WHB - 5 tube failures

The No. 3 tube bundle was replaced with a new tube bundle and the failed unit was sent to NAVSECPHILADIV for analysis. This analysis is currently underway. As part of the failure analysis, several tubes were examined for scale buildup and oxygen corrosion.

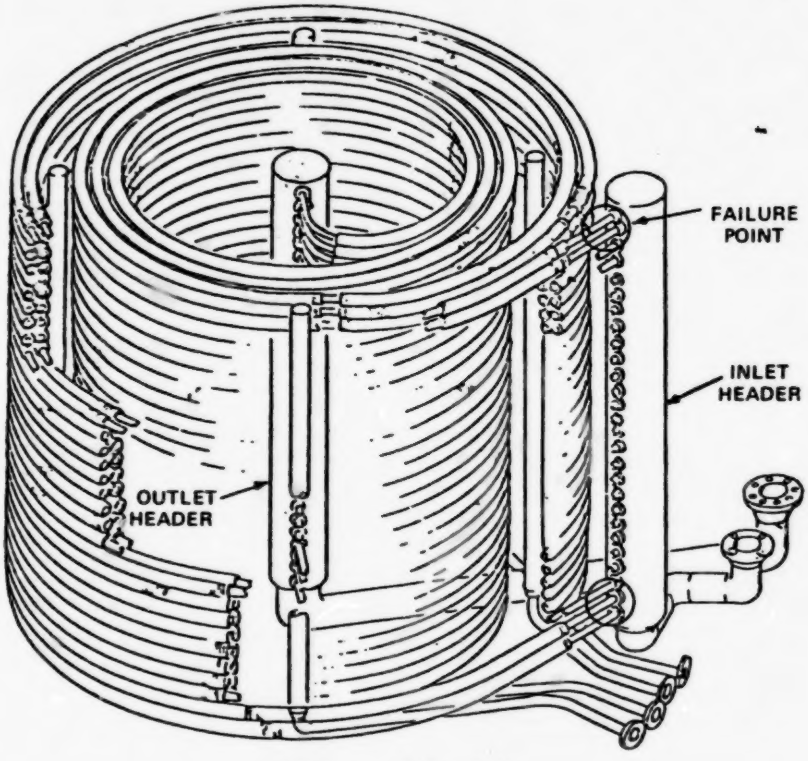

Figure 5. Boiler Coil

A small section of tube showed deposits consisting of iron oxide (See Photograph in Figure 6). Beneath the deposits were large scattered oxygen pits to a depth of 50-60% of wall thickness (See Photograph in Figure 7). By looking back at past problems it is known that the early WHB's were inadequately preserved and oxygen corrosion of tubes was a problem prior to operation on the ship. Also, the units were operated for approximately two years before installing the present treatment system to prevent the problem. A look at the color of the tube deposit shows a brown or black appearance which denoted a passive condition rather than an active condition which would be a red color. It can be concluded that the oxygen corrosion evident in the tube sample occurred prior to installation of new treatment system.

Circulating Pump Seals

The circulating pump is a single-stage centrifugal pump rated at 0.009 m^3/s *(144 gpm) at 207 kPa (30 psig) and 389 K (240 F). The pump shaft was sealed with plastic/metallic packing rings and supplied with feedwater at 339K (150 F) to cool and lubricate the pump shaft and packing rings. On the first sea trial, problems were encountered with excessive leakage from the stuffing box of the circulating pump. The leakage rate could not be controlled to a reasonable amount. This problem was due to the inaccessibility of the pump as located on the boiler foundation. Installation or adjustment of pump packing was impossible. The pump was modified to use a high temperature mechanical seal and the shipboard units were modified to include this mechanical seal.*

In August 1977, less than a year later, reports were received from the ships of an abnormally high failure incidence of the circulating pump mechanical seals. An investigation of the problem pointed to particulate matter (rust) or high temperature as the possible causes of seal failure. A testing program was set-up at the manufacturer's facility to isolate the cause and correct the problem. The pump was operated with the cooling/flushing water secured. This allowed the seal temperature to go to a water temperature of 433 K (320 F) without failure. Then

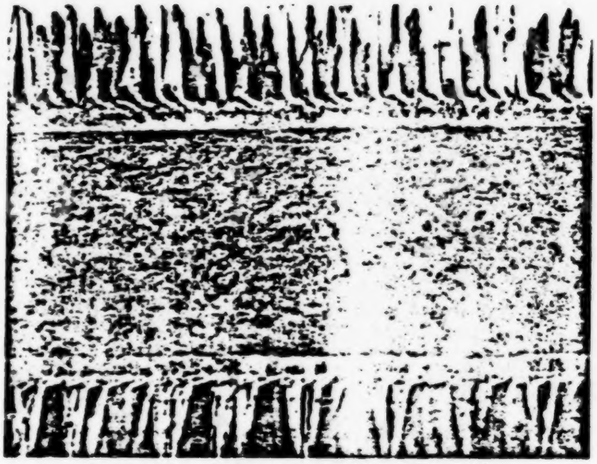

Figure 6. Deposits On Tube Waterside

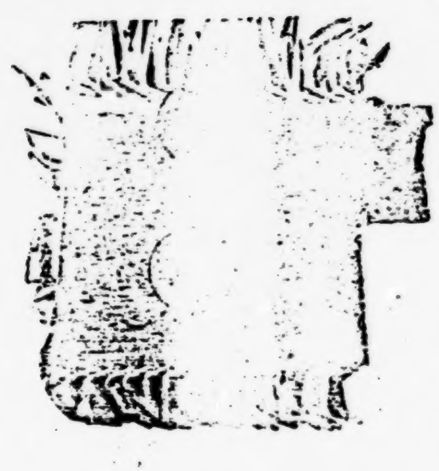

Figure 7. Oxygen Corrosion

rust was injected into the water to simulate shipboard conditions. In a short time the mechanical seal began to show evidence of leakage as the rust deteriorated the seal face. A centrifugal separator was installed in the cooling water line to keep the seal clean of contamination and satisfactory seal performance was obtained. The modification has been backfitted on all shipboard circulating pumps.

Steam Separator

A steam system problem of wet steam was experienced during the first trials of USS SPRUANCE that resulted in malfunction of the distilling plants. As a result, additional drains were added to the steam system to remove condensation during light-off from all low points in the piping system. In addition, an investigation was conducted to determine satisfactory operation of the steam separator to determine if water was being carried-over into the steam system during operation. Testing by the manufacturer revelaed that steam quality was acceptable (99% specification requirement) at design rated condition but dropped off rapidly as the gas inlet temperature was varied beyond the 561-672 K (550-750 F) range. The following modifications to the steam separator were made:

1. *A steam dry pipe was installed in the top discharge connection of the steam separator extending 0.2 m (8 inches) down into the separator.*

2. *A nozzle orifice was installed in the separator inlet to increase velocity in the separator helix.*

3. *A small plate was welded over the top of the separator nozzle and helix to prevent carry-over during high capacity conditions.*

Subsequent tests showed that these changes resulted in full range operation from 533-811 K (500-1000 F) gas inlet temperature with steam quality above 99%. These modifications have been made to all the boilers and have eliminated malfunctions of the distilling plants due to wet steam.

System Stability

A problem of system or control instability has been a continuing problem with the WHB. Fluctuations of the water level in the steam separator were a problem on the first ship trial. The water level in the separator is sensed by a level controller that regulates the feedwater control valve. The volume chambers and direct sensing lines used in the system could not maintain a stable signal to the control valve due to contamination entering the differential controller. A modification was made to the level controls to use bellows and sealed sensing lines. Improvement in the water level control was achieved when this modification was made to the waste heat boilers.

A second control instability problem involved frequent loss of suction by the feedwater pump. The original installation used manual control of seawater flow through the control condenser. As the steam load to the condenser was increased, an increase was required in cooling water flow. If the shipboard operator failed to make this required change, an increased condensate temperature beyond the nominal 328-333 K (130-140 F) range and high feedwater temperatures resulted which caused flashing of water and instability of the feedwater pump. The seawater piping was modified to include a temperature regulating valve that automatically regulates seawater flow to maintain a constant condensate temperature of 333 K (140 F). This modification has been successful.

Drain and Vents

As discussed previously, problems were encountered with wet steam during the early trials. It was found at this time that inadequate traps and drains were installed in the steam system to remove the condensate during initial light-off of the system. A drain (blowdown) connection was needed on the bottom of the inner and outer headers of the boiler coils to permit removal of accumulated sludge. The control condenser and condensate cooler were not fitted with vents from the shell. The non-condensible gases in the system will collect at these two points when the system is lit-off and cause a loss of cooling (condensing) capacity. A vent was needed at the top of boiler headers to remove air from the boiler coils when starting the WHB. These types of deficiencies may seem minor but consider some of the consequences. When the wet steam from the boiler goes to the air ejectors on the distilling plant, this causes a malfunction and shutdown of distilling plants. It may take the operator several hours to get the distilling plant back on the line. In the meantime, the supply of makeup water for the boilers has also been shutdown since the distilling plant is the source of makeup water. The lack of adequate blowdown connections on the boiler will lead to scale buildup with loss of capacity and eventual corrosion problems on watersides.

Accessibility/Maintainability

The boiler and most of the ancillary equipment are located on a common base (see Figure 8). This makes a compact unit which is an advantage for shipboard installation and operation but poses problems for maintenance and/or repair. As originally designed, a quarter panel of the boiler casing was made removable to gain access to the inlet manifold for inspection of boiler watersides and for identifying and plugging leaking tubes. The shipboard installation has a major interference with piping and limited overhead clearance to remove the panel. A modified panel of a smaller size has been designed by the manufacturer which will permit access to the inlet manifold. Included

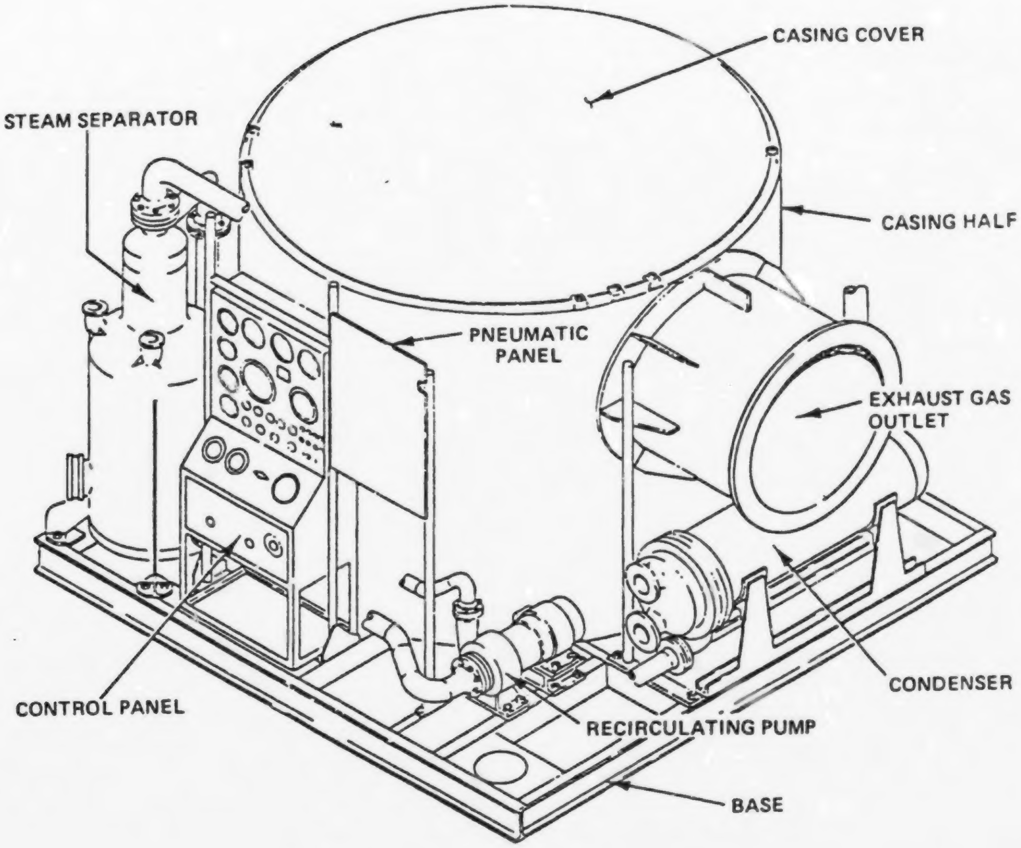

Figure 8. Waste Heat Boiler Arrangement

in the modification is a hinged-mounted boiler control panel that will swing out to improve accessibility. A second problem is that the design configuration of the tube bundle does not permit individual tube replacement. Using the 10 percent rule of thumb, up to 3 of the 27 tubes can be plugged without degradation of steaming capacity. Beyond this, tube replacement is normally required. However, the tube coils are arranged so that each coil supports the coils above it. Removal can only be made starting at the top and working down. Also, the tube coils are too large in diameter to pass through a shipboard hatch. A tube failure problem, such as experienced on USS SPRUANCE, requires a complete bundle replacement through an access hole which must be cut in the ship's structure.

Some other changes that have been made to improve accessibility for maintenance and repair are take-down joints on piping to the circulating pump, relocation of blowdown valves and boiler water sample valves. The No. 3 boiler has many additional problems because this unit was installed in a compartment with extremely limited space around the boiler. A bulkhead has been relocated to increase accessibility to this boiler. Modifications have been made on several ships to remove some of the equipment from the common base to provide improved access to the boiler.

The preventive maintenance program developed for the waste heat boiler has a requirement for inspection of watersides at intervals of 1800 hours of operation and mechanical cleaning if necessary. The problems with the water treatment would generally cause sufficient waterside deposits to require cleaning. As previously seen from the photograph of tube sample, the watersides should have been cleaned. The configuration of the boiler does not permit sufficient access to conduct a waterside inspection, a mechanical cleaning equipment was not provided by the manufacturer or the shipbuilder. The extent of this problem was demonstrated in early 1978 when several tubes on one boiler on USS RADFORD were found to be plugged solid. Several types of mechanical cleaning equipment were used to try to remove the tube blockage, but they could not travel through the complete length of the 25.9 m (85 foot) long tube coil. Chemical cleaning was also tried. However with chemical cleaning, a flow of the chemical solution must be maintained through the tube coil to keep a fresh solution in contact with the deposit. The solution rapidly becomes passive when a fluid flow cannot be maintained and proved ineffective in dissolving the deposit. The manufacturer has developed a mechanical cleaning unit, similar to a "Roto-Rooter", that drives a cable into and out of the tube coil. A brush or cutter is attached in the front end of the cable to remove soft deposits from the tubes. It is necessary to have mechanical cleaning equipment for these boilers. At the operating conditions of the boiler, most deposits that build up on the tubes will be soft deposits of sludge that are best cleaned by a mechanical method.

LESSONS LEARNED

From the previous discussion it can be seen that the waste heat recovery system on the DD-963 class ships has experienced some problems. Changes have been made to correct some of the problems through the concerted efforts of the manufacturer, Conseco; the shipbuilder, Ingalls; and the Navy. Each of these three parties has increased their knowledge and experience as a result of this effort. The development of new equipment for shipboard use as well as improving existing equipment is best achieved by a combined Navy and Industry effort. However, there are specific responsibilities that the Navy and Industry must each assume to make the partnership function efficiently and to provide a good end product. The Navy should provide the definitive specifications for the manufacturer to produce the equipment and the Navy should conduct whatever test evaluation is required prior to accepting the equipment. The experience with the waste heat recovery system on the DD-963 Class Ships has shown that improvements are needed in both of these areas. A third "Lesson to be Learned" is how to test the equipment under realistic conditions. Each of these three issues will be discussed further.

The first issue to be considered is the need for definitive equipment specifications. It is often the case when military specifications do not exist, standard commercial equipment is procured. This equipment is then installed in a shipboard machinery space that has a hot, humid and salt laden environment and is constantly in motion. Most standard commercial grade equipment will not last long in this situation. When the shipbuilding specifications for the DD-963 class ships were developed, no military specifications existed for a waste heat recovery system or boiler, since this was the first time a waste heat recovery system was to be used on a naval ship. As a result an industrial type boiler with slight modifications and some Navy approved components was procured. The results were not good. At this point in time succeeding ships are in the process of being contracted for and built with waste heat recovery systems similar to the DD-963 Class ships. The shipbuilding specifications for the WHB have been re-written by the Navy to include improved requirements based on many of the problems that are described in this paper, but it is a performance type specification. We are relying on the shipbuilder to develop a hardware type specification for procurement of the waste heat boiler. This may result in a satisfactory end product if the same shipbuilder and manufacturer are selected, since much of the experiences lies with these two parties.

However, the Navy should insure a good end product is received regardless of the shipbuilders or manufacturers. The Navy should develop a Military Specification for waste heat recovery systems. Military Specifications exist for propulsion and auxiliary boilers that have evolved through the years and represent the cumulative experience and knowledge of the Navy. These specifications address the material requirements, the design philosophies and requirements, the specific measurable testing requirements, the preservation and packaging requirements, and the quality assurance requirements. These requirements can be coupled with the lessons learned from the DD-963 Class shipboard experience to develop a Military specification for the waste heat recovery system that will ensure a quality product for all future ships.

A second consideration is where to conduct the acceptance test program. Testing which is intended to evaluate the applicability and worth of a product proposed for U.S. Navy use is the responsibility of the Navy. To state that the manufacturer is going to be less objective in conducting an evaluation of his own product raises a sensitive issue. However, it is clear that putting a manufacturer in such a position creates a "conflict of interest" potential.

An important factor is the knowledge of the shipboard environment including the capability of the ultimate user which a testing activity must consider in evaluating a product. The evaluation by the manufacturer, of the performance of his product, when operated by his employees, in his plant can be significantly different from the evaluation of experienced Naval engineers of a product subjected to testing under shipboard operating conditions and with knowledge of the capabilities and limitations of the shipboard user. It might be argued that a Manufacturer conducted test with Navy on-site monitoring would eliminate such a problem. In my opinion and experience this is not entirely true. There are many lessons learned and potential deficiencies uncovered by the installers, operators and maintainers in the course of a machinery test program. These numerous detailed points are passed up thru shop supervision to test engineers to the test project manager. It is normally at this level that communications take place between Navy monitors and the manufacturers representative. The amount of detailed information passed to the Navy representative is based on the preception of the various manufacturers personnel as to its importance. It is in this process that information and experience is lost. With a Navy test program the personnel performing the installation, operation and maintenance are in direct contact with, and in many cases, are the same people who will be making the evaluation judgement and who will be provideing follow-on assistance to the fleet for the system. Thru this process all experience is brought to bear in the evaluation and retained and utilized by Navy personnel.

With the DD-963 Class ships, the Navy has the long term responsibility for the waste heat recovery system but to date Conseco and Ingalls have the majority of the experience and knowledge. They may not be available at some time in the future as they have been in the past to assist in problem resolution. In-house testing at a Navy facility affords the opportunity for Navy engineers and technicians to become familiar with new systems such as a waste heat recovery system, at the earliest possible time. Performance of such testing provides design understanding and baseline-condition knowledge that is invaluable throughout the life of the shipboard system. When the system goes to sea, there is a fully trained staff of Navy personnel available to assist the Fleet immediately and for years to come.

The third issue to be considered is how to test equipment. The DD-963 waste heat boilers were tested by the manufacturer with a combustion test rig. The mass flow and temperature of the gas can be simulated with the test rig, however, the parameters of pressure, velocity profiles, and gas combustion constitutents cannot be easily simulated. When the boilers were installed on the ship a two fold increase occurred in the gas side pressure drop due to the actual engine. This resulted in a decrease in system performance. Secondly, the soot blowers were found to be ineffective in cleaning the gas sides of the boilers of the accumulated soot from the engine. Again this results in a decrease in system performance. The problem of chemical hideout could be a result *of the turbulent gas flow from the engine causing localized overheating or hot spots on some tube areas. These three problems were not identified during the qualification tests and are still not resolved at this time. An additional design feature that should be demonstrated is the requirement to integrate the boiler controls with the gas turbine controls to achieve the "push button automation" required. This requirement cannot be verified with a combustion test rig. Some of the control stability problems previously discussed could have been identified and resolved during a test program utilizing a gas turbine generator set. In acceptance testing the question that has to be answered is, "Will the WHRS perform satisfactorily in the shipboard environment in conjunction with the other components it is integrated with?" This can only be accomplished by conducting the WHRS test in conjunction with the gas turbine generator set used on the ship and operated in the same manner as shipboard. An added benefit of the combined system test is the capability to construct a mock-up of the actual shipboard configuration around the system. The mock-up will provide for a realistic maintainability evaluation that will surface any accessibility or maintainability problems prior to installation on shipboard.*

SUMMARY

The DD-963 Class ships are the first U.S. Navy vessels to utilize a waste heat recovery system on a gas turbine engine. The experience with this system after 3 years of service aboard ship has shown that problems do exist. This is to be expected when new equipment is introduced into the Fleet. This experience has been used to develop three areas for consideration that can improve the probability of success in future procurements. They are (1) the need for definitive military specifications; (2) the need to test at Navy Laboratories; and (3) the need to test complete systems under simulated shipboard conditions.

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